STEAM AND OTHER ENGINES

S R Hosangadi
Baroda.

BY

J. DUNCAN, Wh.Ex., M.I.Mech.E.

HEAD OF THE DEPARTMENT OF CIVIL AND MECHANICAL ENGINEERING AT THE MUNICIPAL TECHNICAL INSTITUTE, WEST HAM

AUTHOR OF "APPLIED MECHANICS FOR BEGINNERS"

MACMILLAN AND CO., LIMITED
ST. MARTIN'S STREET, LONDON
1923

First Edition 1907.
Reprinted 1909, 1911, 1913, 1915, 1918, 1919 (twice), 1920 (twice), 1923.

PRINTED IN GREAT BRITAIN

PREFACE.

THE object of this book is to provide students of Engineering with an explanation of the elementary principles of science applicable to heat engines and with a description of the essential constructional details of typical engines. These details have been treated as fully as space permitted, and the plan adopted has been to describe one complete design of each type very fully, so as to enable the student to undertake with confidence the dissection of other kinds of machines.

Though the scope of the book has been in no way limited by the syllabus of any particular examination, the student who works through the following pages will have no difficulty in qualifying in the more elementary examinations of the Board of Education and other examining bodies.

A knowledge of elementary Practical Mathematics and of the fundamental principles of Machine Construction and Drawing has been assumed, and it has been taken for granted, since the subjects assist one another very greatly, that the student will be studying Applied Mechanics at the same time that he undertakes the work dealt with in this book. Prominence has been given in the earlier chapters to an experimental study of Heat, as teachers of Engineering find that it is exceptional for the student to come to his classes with any adequate knowledge of this subject.

Numerous worked-out numerical examples are given in the text, and these are intended as much to inculcate a good style of work as to elucidate the principles dealt with. It is hoped that these exercises will be studied thoroughly and be supplemented by the problems given at the ends of the chapters. Of

these questions those with a date appended are from examination papers of the Board of Education.

The experiments described at each stage of the work should be performed by the student himself and the exercises based upon them carefully worked out. A course of laboratory work has been provided at the end of the volume, and since the equipment of engineering laboratories varies so greatly this has been made as general as possible.

This opportunity is taken gladly to acknowledge various sources of assistance. To Prof. Ewing's valuable work on the Steam Engine the author owes much of his knowledge. For many working drawings and for much practical information, the author is greatly indebted to Mr. James Holden, Locomotive Superintendent of the Great Eastern Railway, and to Mr. Robert Warriner, Manager of the Thames Engineering works. Mr. A. T. Quelch, Assistant Lecturer in Engineering in the West Ham Technical Institute, has kindly read the proofs. Prof. R. A. Gregory and Mr. A. T. Simmons have given the author the benefit of their experience in the production of books for students and have read the proof sheets.

Many firms have permitted the use of their copyright illustrations, and the source of each of these is made in appropriate places. The gas engine illustrations in Chapter XIX. have been made direct from the engine, and thanks are due to Messrs. Crossley Bros. for the privilege of publishing them.

Prof. John Perry, F.R.S., kindly gave permission for the inclusion of the Tables on the properties of steam from his book on "The Steam Engine and Gas and Oil Engines" (Macmillan); the Logarithmic Tables are from Mr. F. Castle's "Machine Construction and Drawing" (Macmillan).

JOHN DUNCAN.

WEST HAM,
July, 1907.

CONTENTS.

STEAM AND OTHER ENGINES.

CHAPTER I.

INTRODUCTION.

Action of simple engines.—Steam is generated from water by the application of heat. When an open vessel is used, the steam is given off at the same pressure as that of the atmosphere, but a much higher pressure may be secured by generating steam in a closed vessel. Steam may be used for the production of work by allowing it to push a piston to and fro in a cylinder; or, by

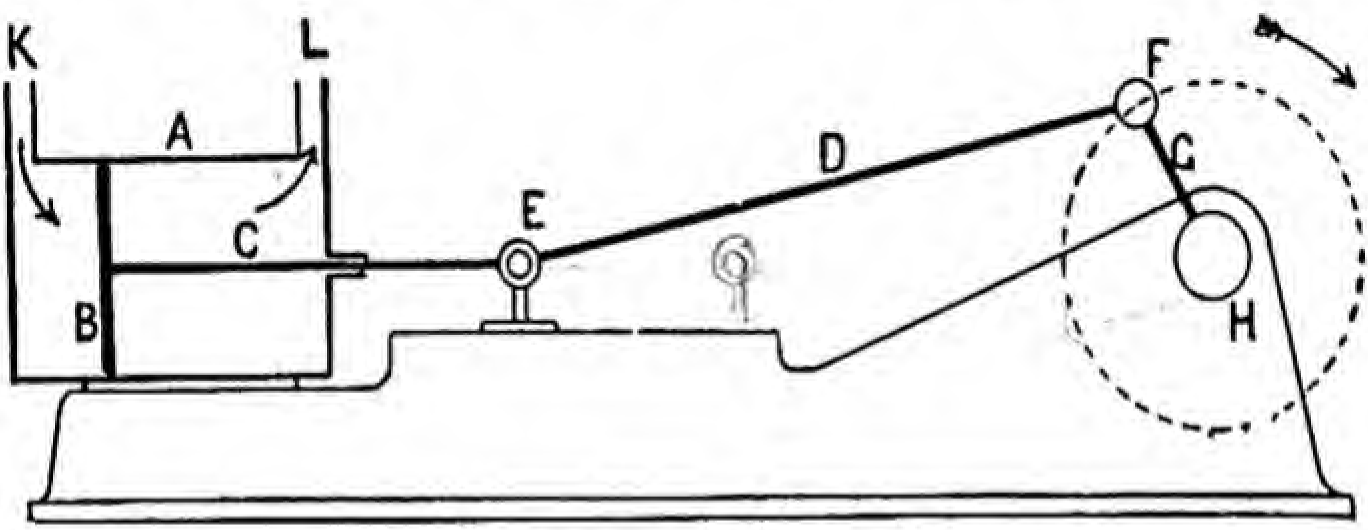


FIG. 1.—Outline diagram of engine mechanism.

causing it to discharge against vanes formed round the circumference of a wheel, thus producing rotation. In this Chapter, sufficient information will be given to enable the student to understand the action and arrangement of the parts of a small engine of the first-mentioned type.

Steam engines having pistons working in cylinders are generally employed to give a motion of rotation to a shaft. This result is effected by means of a mechanism called the crank and connecting rod. In Fig. 1, *A* is a **cylinder** shown in section. *B* is a **piston** capable of sliding in the cylinder and fitted so as to prevent leakage

of steam past its edge. A **piston rod** *C* is attached to the piston and passes through a hole in one end of the cylinder, formed so as to be steam-tight. The outer end *E* of the piston rod is jointed by means of a pin to one end of a **connecting rod** *D*. The other end of the connecting rod is attached to a pin *F* secured to a **crank** *G*, which is mounted on a **crank shaft** *H*. As the shaft *H* rotates, the pin *F* will describe the circumference of a circle, and the pin *E* will move to and fro in a straight line.

Steam is admitted to the cylinder first through the opening or **port** *K* and will exert pressure on the left-hand side of the

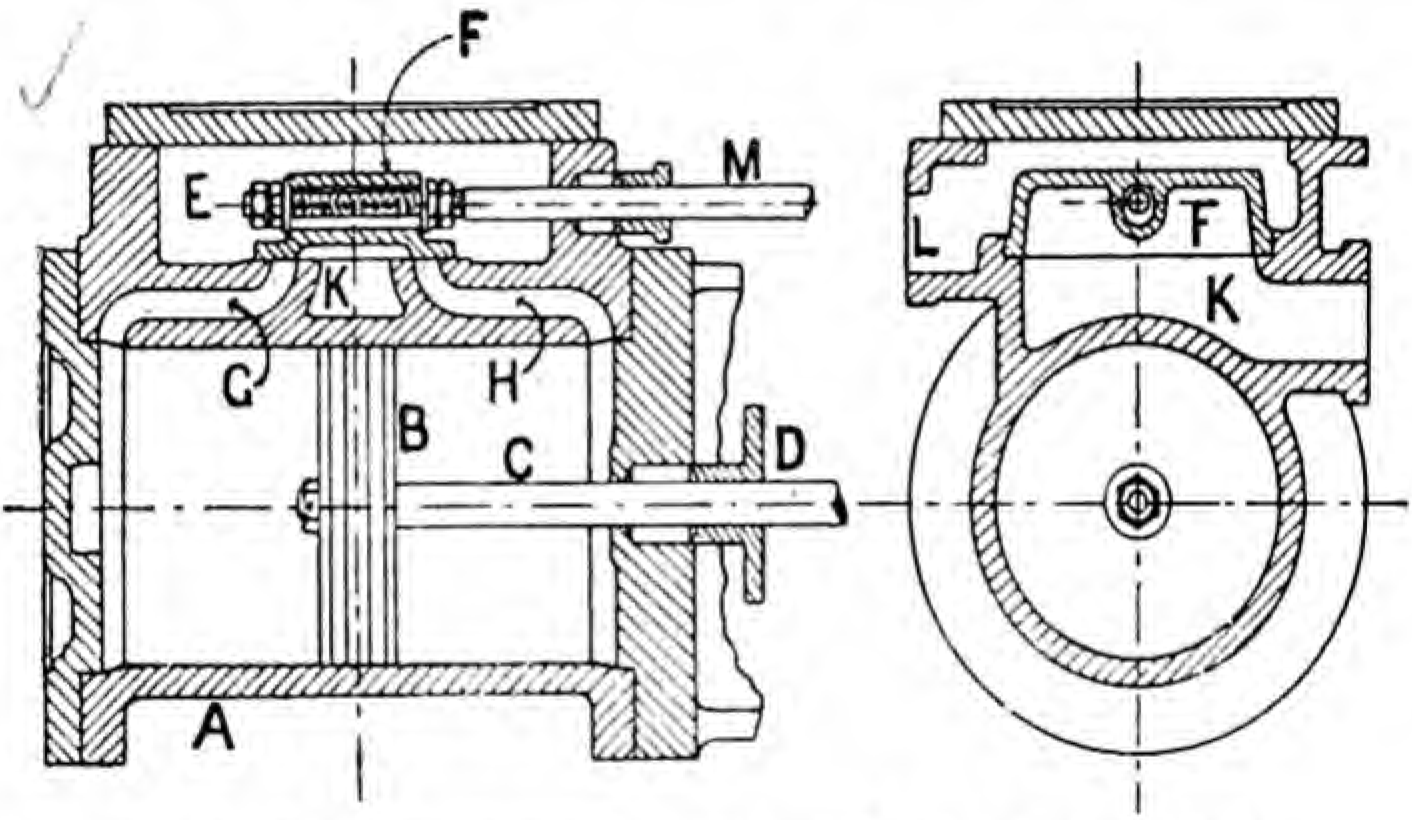


FIG. 2.—Sectional side and end elevations of a steam engine cylinder.

piston, the other side being put into communication, through the port *L*, with the atmosphere or with a vessel called a **condenser**, in which the pressure is kept low. The steam, by its pressure, will cause the piston to travel to the right-hand end of the cylinder, and thus, by means of the intervening mechanism, the crank shaft is made to execute half a revolution. Steam is then directed into the right-hand side of the cylinder through *L*, the left hand portion being put into communication with the atmosphere or with the condenser through *K*, and the piston will be driven back to the left-hand end of the cylinder, the crank shaft meanwhile completing the revolution. To enable the crank shaft to rotate smoothly without jerky action, a heavy wheel called a **fly-wheel** may be attached to it.

Construction of the cylinder.—The cylinder is constructed with various passages, or ports, formed in it so that the steam may flow as required. The distribution of the steam first to one side and then to the other side of the piston is effected by means of **valves**, of which there are many varieties. These valves are opened and closed at the proper instants by means of a mechanism driven from the crank shaft, so that the engine is self-acting.

A common form of cylinder is shown in section in Fig. 2. *A* is the cylinder, made of cast-iron, and fitted with a cast-iron piston *B*. *C* is the piston rod, secured firmly to the piston, and passing through a hole *D* in the cylinder end. A box *E* of rectangular shape, called the **steam chest**, is cast on to the side of the cylinder, and is provided with a movable cover. Two steam ports *G*, *H*, lead from the steam chest, one to each end of the cylinder. A third port *K*, called the **exhaust port**, leads from *E* into the atmosphere, or condenser. These ports open into the steam chest at a flat face (Fig. 4) over which a valve *F* (Fig. 2) is arranged to slide to and fro, being driven by a rod *M* actuated by mechanism mounted on the crank shaft. Steam is brought from the boiler into the steam chest through an opening *L*, and is distributed by means of the **slide valve** *F*. This valve is made like a rectangular box turned upside down (Fig. 3).

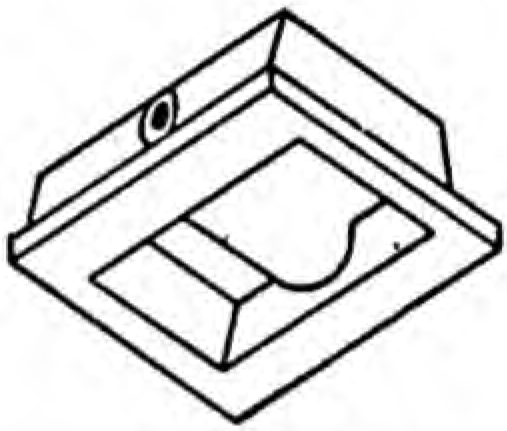


FIG. 3.—Perspective view of a slide valve.

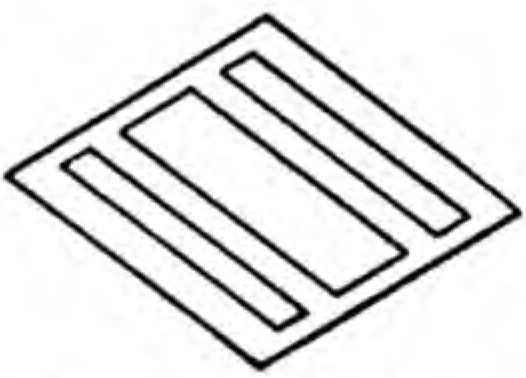
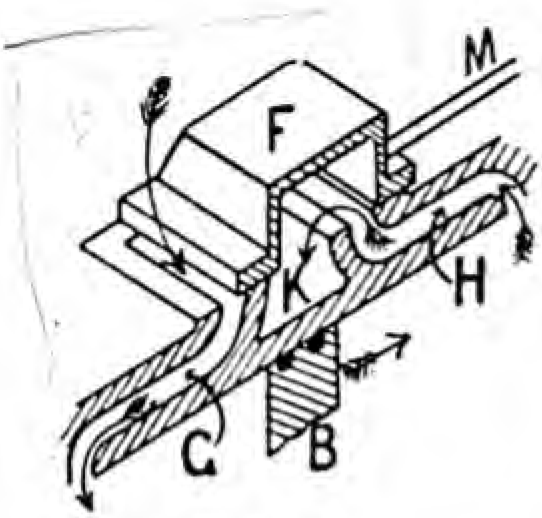


FIG. 4.—Perspective view of the cylinder face and ports.

Action of the valve.—To understand the action of the valve reference should be made to Fig. 5, in which a perspective view of the valve and ports is shown in section. As the valve is positioned in this figure, the steam may flow from the steam chest through *G* into the left-hand part of the cylinder. The other side of the piston is in communication with the exhaust port *K* through the port *H* and the cavity in the valve. The piston *B* will thus be caused to travel towards the

right. The return stroke of the piston is effected by first moving the valve into the position shown in Fig. 6. The steam will now be directed through the port H into the right-hand portion of the cylinder, the other side at the same time being put into communication with the exhaust through the port G and the cavity of the valve. Motion of the piston towards the left will



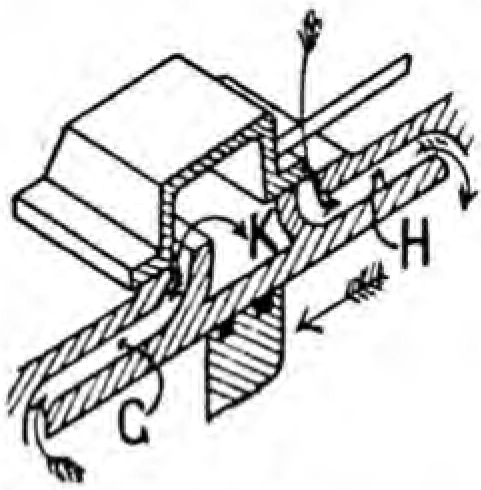


FIG. 5. FIG. 6.

Diagrams showing the way in which the slide valve distributes the steam.

now occur. The valve is generally designed so as to admit steam to the cylinder only during the early part of the stroke, and then to cut off the supply, the remaining part of the stroke being accomplished by the **expansive action** of the steam, giving a continually diminishing pressure on the piston. This arrangement is adopted as being more economical.

Method of driving the valve.—The valve is driven by means of a device called an **eccentric**, which consists (Fig. 7) of a

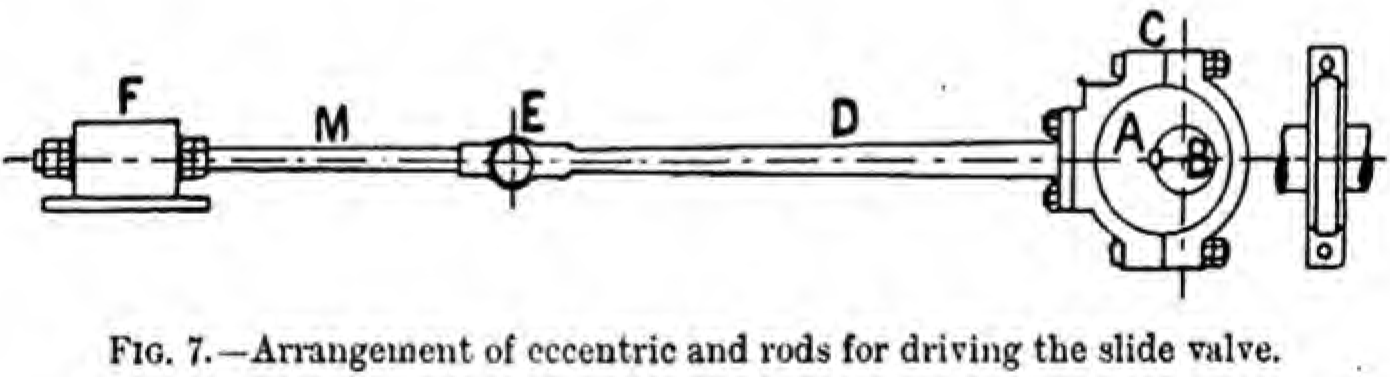


FIG. 7.—Arrangement of eccentric and rods for driving the slide valve.

circular disc A, having a hole bored through it to receive the crank shaft B. The hole is bored a short distance away from the centre of the disc, and a key is fitted so as to secure the disc firmly to the shaft. The disc is called the **eccentric sheave**; its edge is made to receive a **strap** C, which is a working fit on the

sheave in order that the sheave may rotate with the crank shaft without producing motion of rotation in the strap. The strap is made in halves, secured together by means of bolts, so that it may be got into position round the sheave. Attached to the strap by means of studs is an eccentric rod *D*, the other end of which is attached by means of a pin at *E* to the end of the valve rod *M*. As the crank shaft rotates, the valve will be driven to and fro a distance equal to twice the distance from the centre of the eccentric sheave to the centre of the crank shaft. As will be explained later, the whole mechanism is set so that the ports are opened or closed, to steam supply or exhaust, at the proper times.

Construction of the piston.—Pistons for small engines are generally made of cast-iron. Forged steel, or cast steel, is more suitable for pistons having a large diameter. There are many different ways of attaching the piston rod to the piston, the principle to be borne in mind being that, owing to the alternate push and pull on the rod when the engine is at work, there is a risk of the piston becoming slack on the rod. A common method is to taper the end of the rod to fit a tapering hole in the piston (Fig. 8). The rod is then pulled home by means of a nut fitting a screw on the rod. The nut is prevented from slacking back by means of a split pin passing through both the nut and the rod.

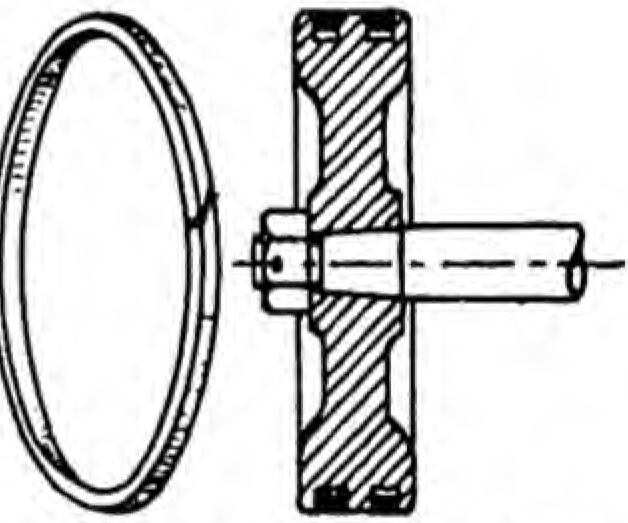


FIG. 8.—Piston and spring packing ring for a small steam engine.

The piston is made steam-tight in the cylinder by means of **spring rings** fitting into grooves turned in the rim of the piston. For small pistons, the arrangement shown in Fig. 8 suffices. There are two rings, usually of cast-iron, of rectangular section or nearly so. These are turned of diameter rather larger than the cylinder, then split at one place and sufficient material removed to allow of the rings being sprung into place in the cylinder; their edges will then meet. The rings are first sprung over the rim of the piston into the grooves formed to receive them, and the piston

is then pushed into the cylinder, when the rings will press outwards against the cylinder walls and so prevent leakage of steam past the piston.

Stuffing-box and gland.—To enable the piston rod and valve rod to pass through the cylinder end and the steam chest respectively without leakage of steam occurring, stuffing-boxes and

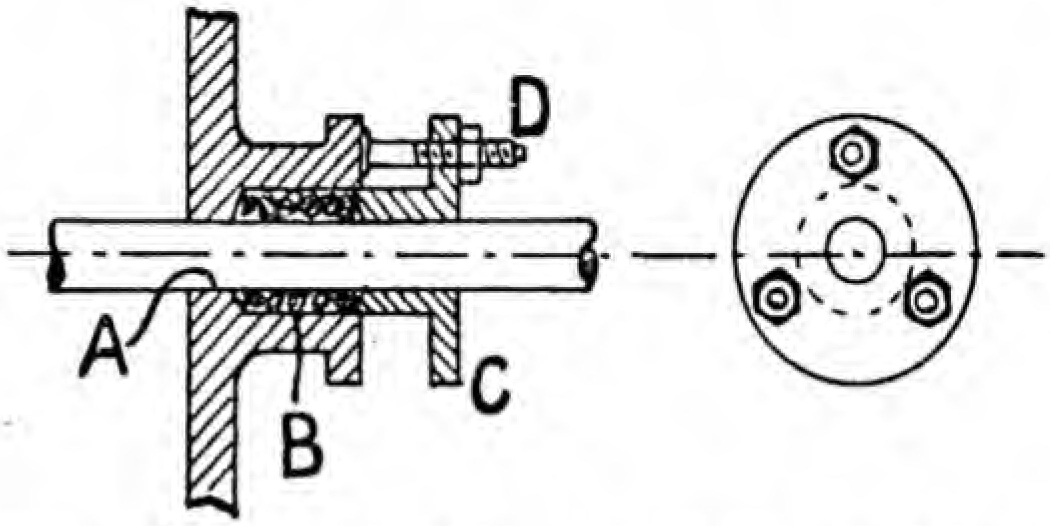


FIG. 9.—Stuffing box and gland for rendering the piston rod steam-tight.

glands are fitted. Fig. 9 shows the arrangement in section. The hole is bored out a working fit to receive the rod at *A* near its inner end. The outer part *B* is enlarged to receive packing, which may consist of asbestos or some similar substance. The packing is held in position and forced home by means of a gland *C* and studs *D*. On screwing down, the packing is pressed against the walls of the stuffing-box and also against the rod and so prevents steam leakage.

Attachment of the valve rod.—The valve rod must be attached to the valve in such a manner as to prevent lost motion and at

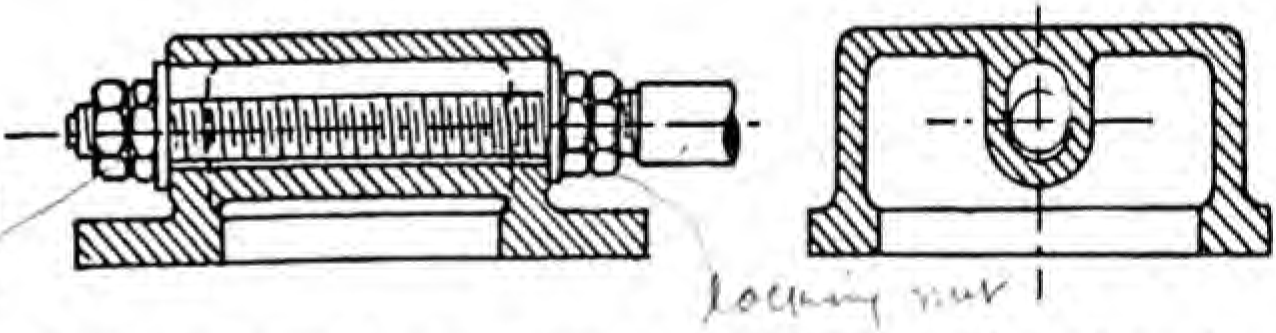


FIG. 10.—Sections of a slide valve, showing a method of attaching the valve rod.

the same time to permit of the valve being pushed firmly against the cylinder face by the steam pressure. The latter condition may

be secured by making the hole through the valve for receiving the rod oval in the direction perpendicular to the cylinder face (Fig. 10). Locking nuts are fitted to the valve rod on each side of the valve; these may be adjusted so as to remove any slackness in the direction of motion of the valve and not tightly enough to bind the valve on the rod and so prevent the valve being pressed against the cylinder face.

Cylinder drains.—Water in considerable quantities, produced by the condensation of some of the steam, is apt to be present in the cylinder. This is always the case at first starting, and means must be provided for getting rid of the water, otherwise there is risk of there being enough present to fill the whole of the clearance space between the piston and the cylinder covers when the piston is at the end of the stroke. Should this occur, the cylinder covers may be knocked off or burst by the returning piston. Fig. 11 shows an arrangement of two **drain cocks** *A A*, screwed into holes near the ends of the cylinder at its lowest side. The handles of these cocks are connected and may be operated by the handle *B*. When the cocks are opened, a free exit is provided for water to escape through the drain pipes *C*.

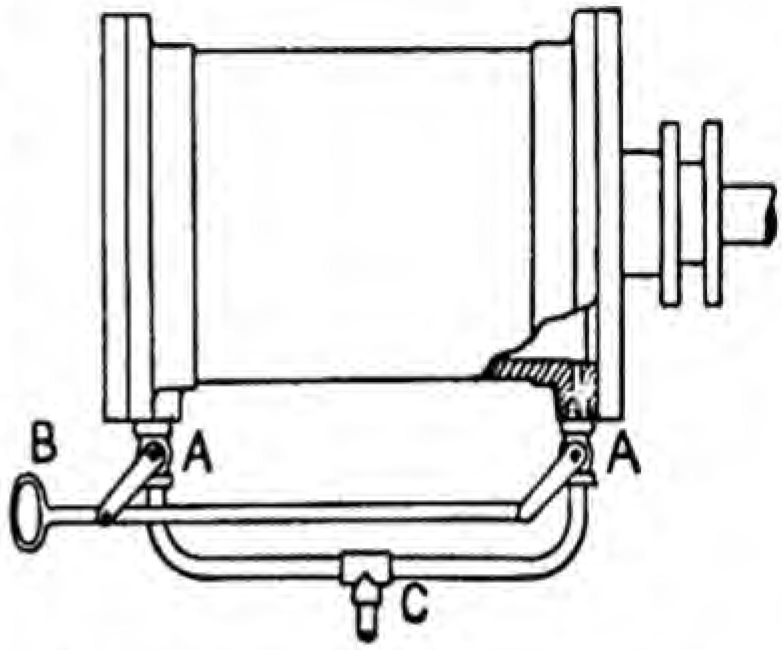


FIG. 11.—Arrangement of cylinder drains.

Crosshead and guides.—The arrangement by means of which the connecting rod and the outer end of the piston rod are connected is called the **crosshead**. A type of crosshead suitable for a small engine is shown in Fig. 12. This crosshead consists of a block *A* having brass bushes *B* fitted so as to form a bearing for the **crosshead pin**, which in this type is secured to the connecting rod (*B*, Fig. 14). The bushes are held in place by a cap *C* and studs provided with locking nuts. The end of the piston rod *D* is tapered to fit a corresponding hole in the crosshead, and the rod is pulled home by means of a **cotter** *E*. In order to guide the end

of the piston rod, and so to counteract the oblique action of the connecting rod which tends to bend the piston rod, the lower part of the crosshead (called the **slipper**) is formed as shown in Fig. 12.

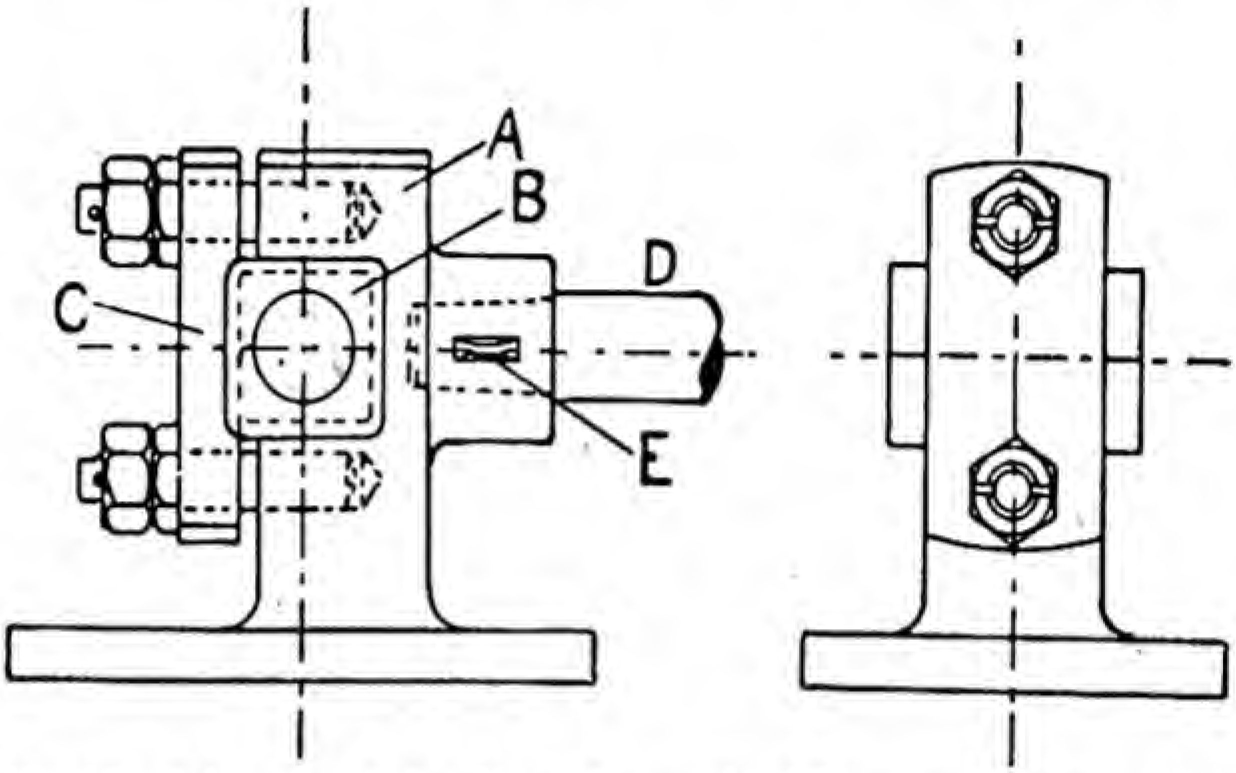


FIG. 12.—Side and end elevations of a crosshead for a small steam engine.

The flat face at the bottom of the slipper slides on a planed surface forming part of the frame of the engine. One arrangement of **guide** for the crosshead is shown in Fig. 13, where A is the planed surface and BB are **guide bars** held down by studs.

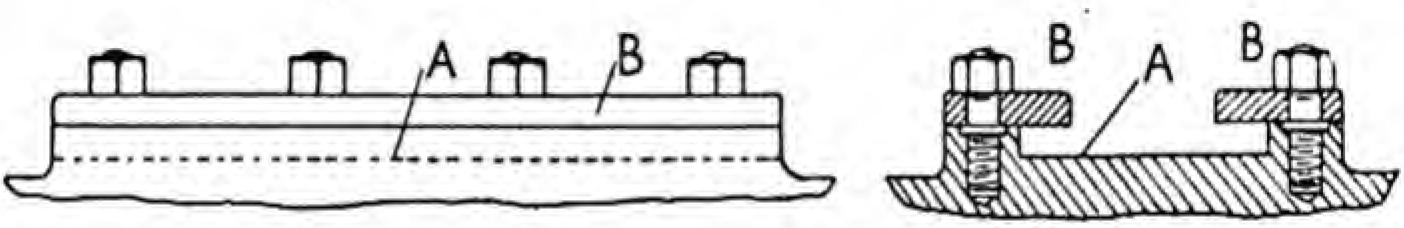


FIG. 13.—Side elevation and section of guides.

The slipper is thus controlled in all directions except in that of the movement of the end of the piston rod, which accordingly is compelled to travel in a straight line.

Connecting rod.—In Fig. 14 a common form of connecting rod is shown. The rod is forked at A so as to embrace the crosshead. B is the crosshead pin, which is sometimes shrunk in, or, as is shown in Fig. 14, secured by two taper pins. The other end of the rod is palmed at C, so as to serve for the attachment of brass bushes D which form the bearing for the crank pin. The bushes are secured by means of a cap E and two fitted bolts, with lock

nuts and split pins to prevent slacking back. The connecting rod tapers slightly from the crank pin end to the crosshead end.

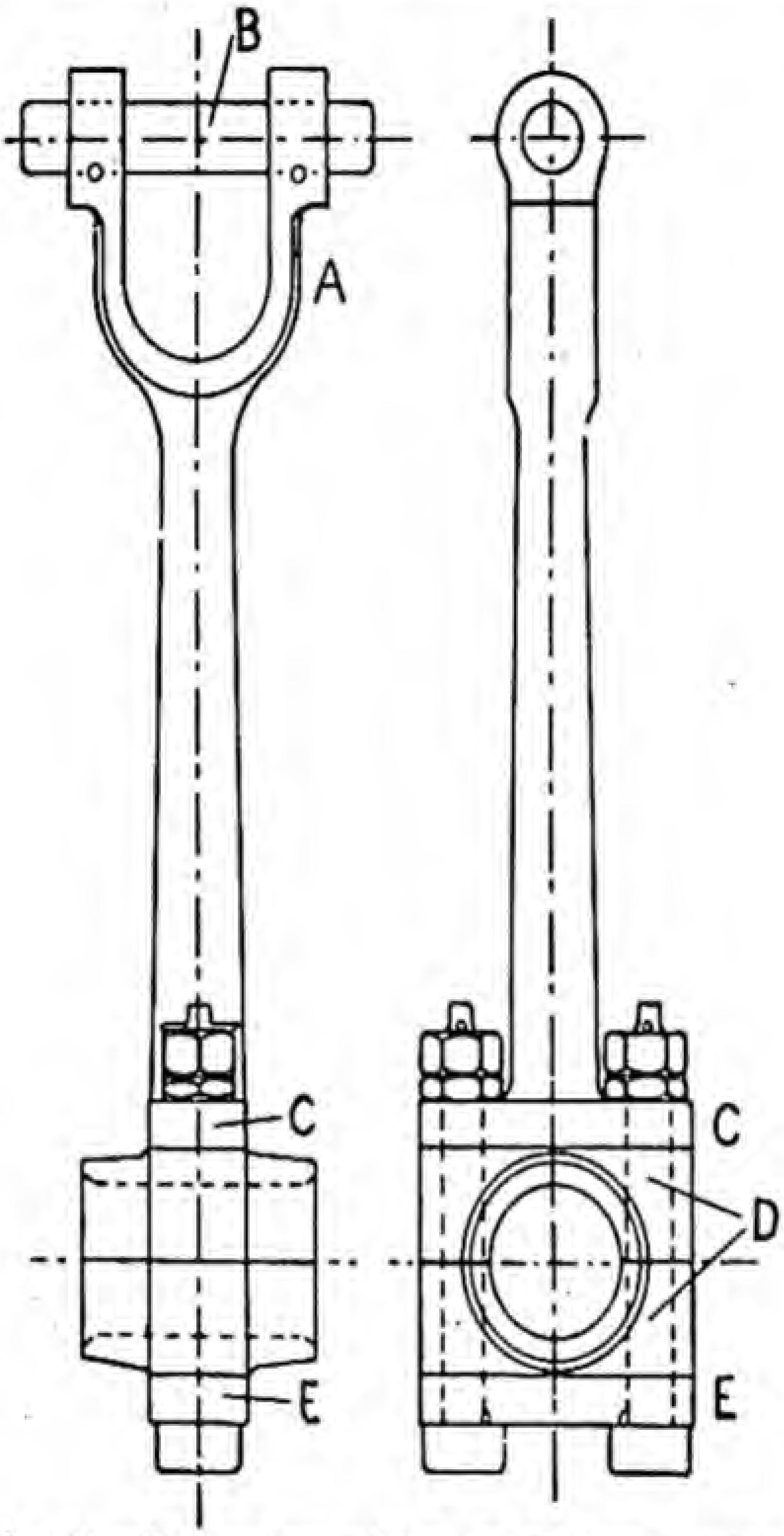


FIG. 14.—Connecting rod for a small steam engine.

Crank shaft.—Crank shafts are sometimes cut out of the solid, others are built up; occasionally they are bent from an initially

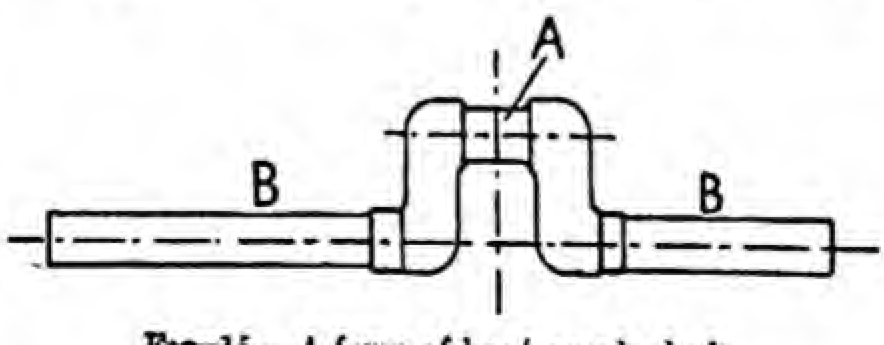


FIG. 15.—A form of bent crank shaft.

straight bar. Fig. 15 illustrates one of the latter type. The crank pin is turned at *A* and the straight portions *BB* are also

turned to receive the **main bearings**, the fly-wheel and the eccentric. A form of main bearing, which is the name given to the bearings in which the crank shaft rotates, is shown in Fig. 16. Both views

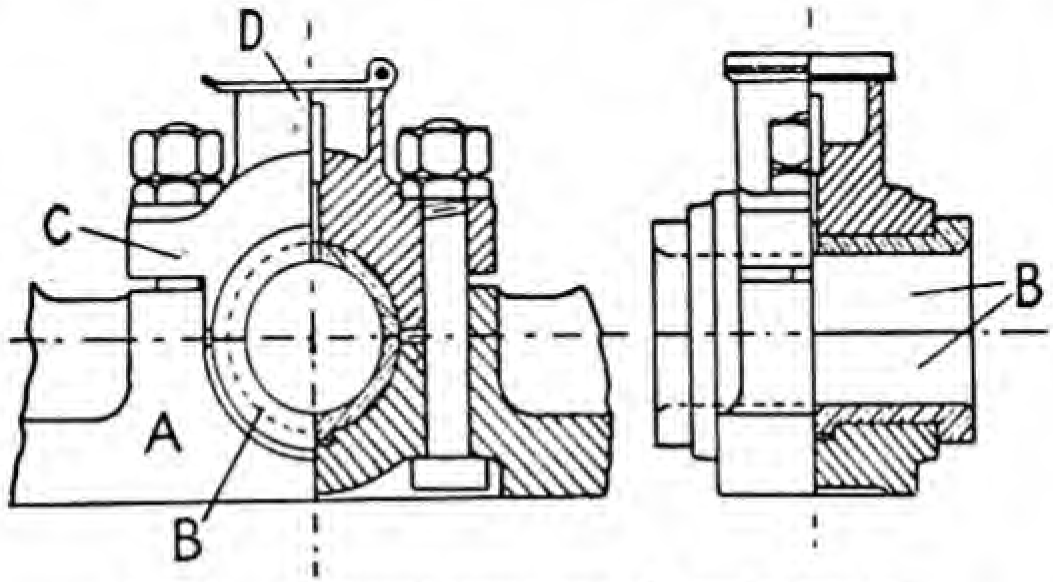


FIG. 16.—Sectional elevations of a main bearing.

are partly in section in order to show the construction. The engine frame is here shaped as shown at *A*, and is made to receive brass bushes *B* held down by a cap *C* and two bolts with locking nuts. An oil cup *D* forms part of the cap.

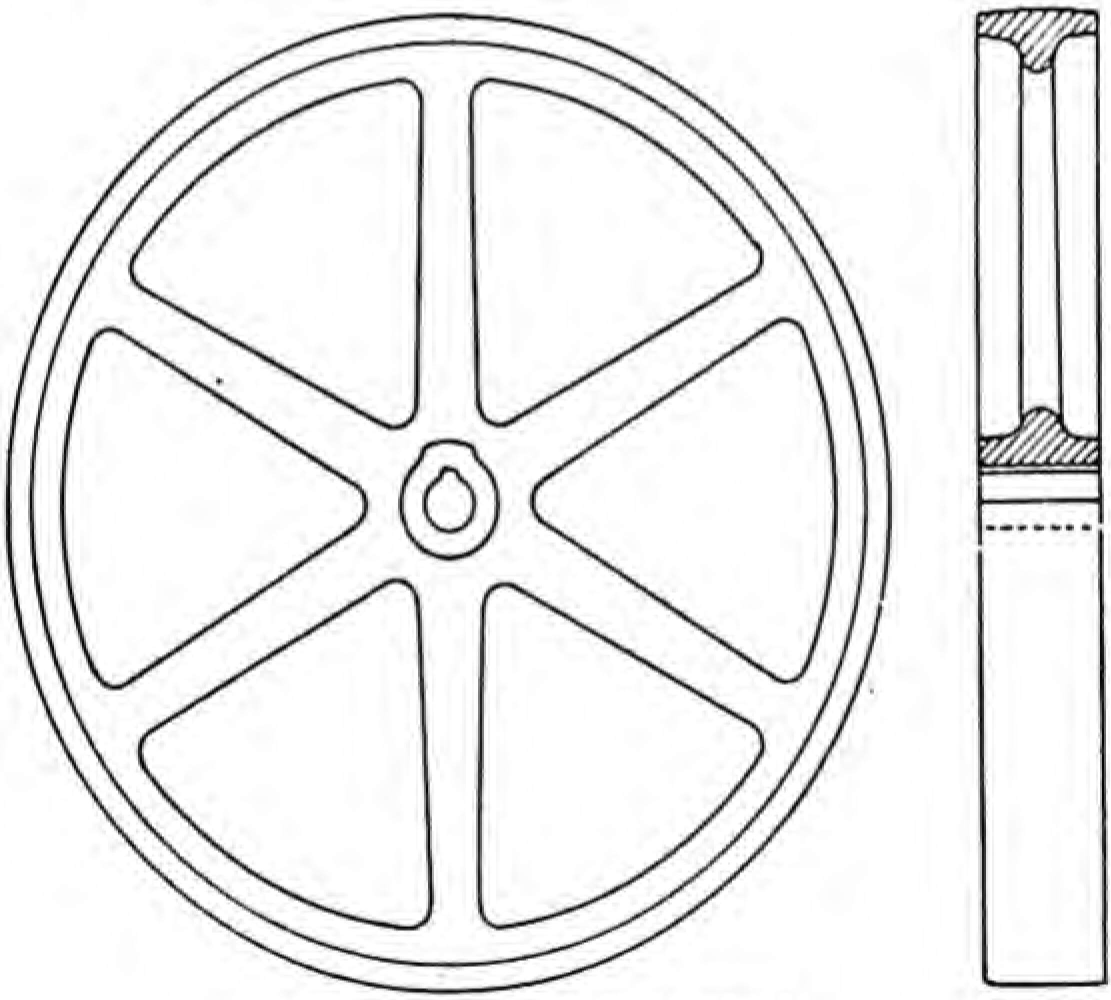


FIG. 17.—A small fly-wheel.

A type of **fly-wheel** suitable for a small engine is shown in Fig. 17; it will be noticed that the wheel is cast in one piece in this

example. The hole at the centre is bored to fit the crank shaft and has a keyway for receiving the key by means of which the wheel is secured to the shaft. The function of the fly-wheel is to produce steady rotation free from jerky action.

The soleplate.—The various parts of the engine are secured to a casting called the **soleplate,** which rests on the engine foundation and is securely bolted thereto. A common pattern of soleplate is shown in Fig. 18. The end *A* forms the front cover of the

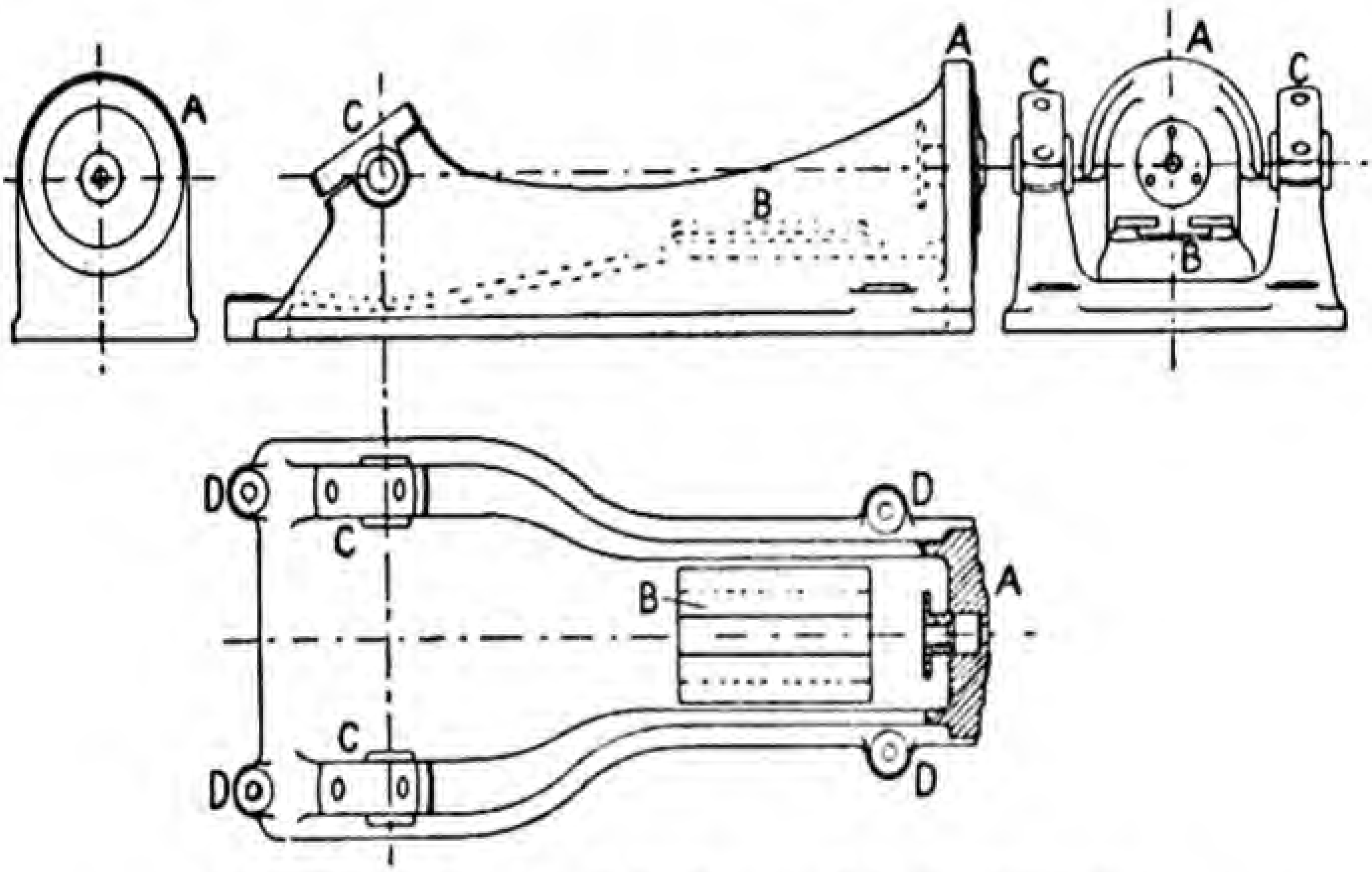


FIG. 18.—A soleplate suitable for a small horizontal engine.

cylinder and is fitted with a stuffing-box for the piston rod. The cylinder is bolted against *A* and overhangs. The guides are formed at *B*. *CC* are the main bearings. Provision is made at the places marked *D* for bolting the soleplate to the foundation. The design of this soleplate requires care, as it receives and transmits considerable forces from the cylinder to the main bearings.

The governor.—The engine is kept at a steady average speed of rotation by means of a device called a **governor.** The governor effects this by regulating the supply of steam to the cylinder. The arrangement is shown in Fig. 19, where *A* is the governor and *B* is the steam pipe supplying steam to the cylinder. A **throttle valve** *C* is placed in the steam pipe; in the example shown, this valve consists of a disc which may be rotated partially on an axis *D* by means of a lever *E*. If the disc is situated transversely to

the pipe, steam will be cut off almost entirely, and there will be a more or less free passage past the valve depending on the angle at which it is set. The function of the governor is to control this angle. Two heavy balls *FF* are mounted at the end of arms *GG*, which are attached by pins to a spindle *H*; the latter is rotated by a belt from the crank shaft running on the pulley *K*, the motion being transmitted to *H* by means of bevel wheels *L*. Other arms *MM* connect the balls to a sleeve *N* which may slide on the spindle *H*. A heavy weight *P* bears downwards on the sleeve. A bent lever *Q* has one arm connected to a collar embracing the sleeve and the other arm is connected by a rod *S* to the throttle valve lever *E*.

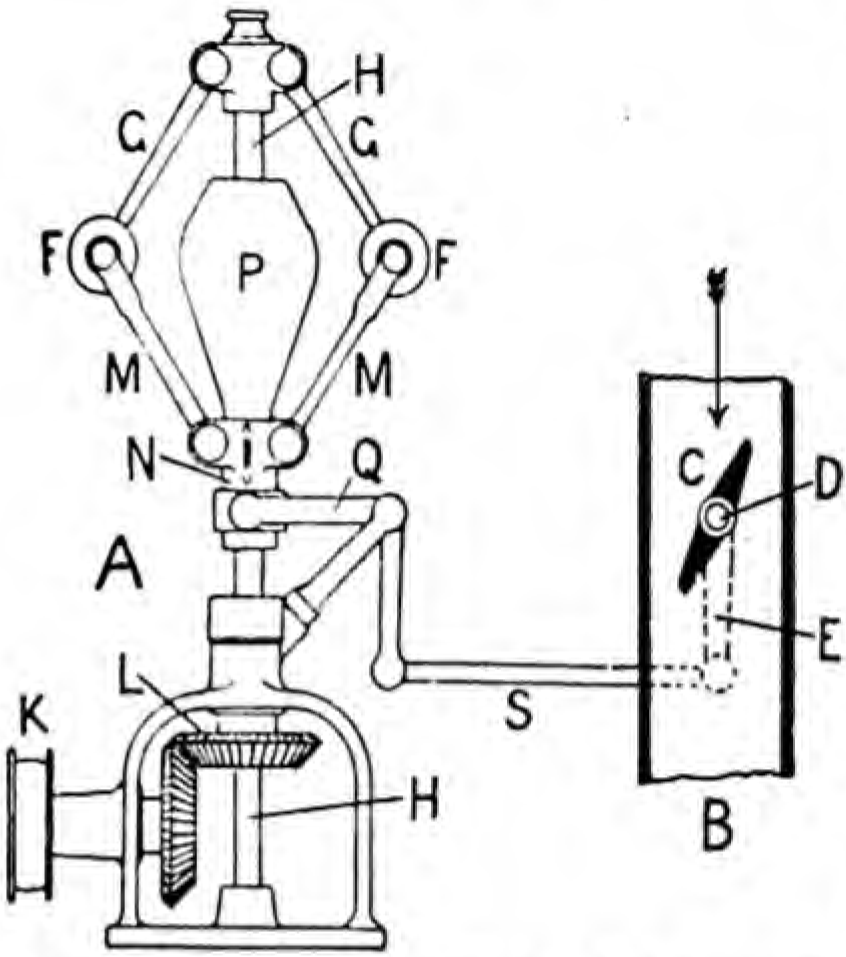


FIG. 19.—Diagram showing how the governor controls the steam supply.

When the engine is running and driving the governor spindle, centrifugal force acts on the balls, causing them to move outwards until a steady position is reached which depends on the speed of rotation. Any increase in the speed will cause further outward movement of the balls, producing an upward movement of the sleeve *N*, and this, being transmitted through the levers and rod to the throttle valve, will close it partially, and so reduce the supply of steam to the engine; the engine speed will thus fall. Should the speed be lowered below the proper amount, the balls move inward, thus lowering the sleeve *N* and so opening the throttle valve more and thus permitting more steam to pass to the engine.

Arrangement of small steam plant.—The student will now find it instructive to examine carefully the drawings of a small steam engine and boiler shown in Figs. 20 and 21, and reproduced here by the courtesy of Messrs. E. S. Hindley & Sons. The engine

is of the **horizontal type**, *i.e.* the centre line of the cylinder is horizontal. The steam cylinder is shown in section at *A* (Fig. 21); the cylinder is bolted to the end of the soleplate *B*, which is of a similar type to that illustrated in Fig. 18. The crank shaft *C* is supported by main bearings *DD*, and has mounted on it a flywheel *E*, an eccentric *F* for driving the slide valve, and a belt pulley *G* for driving the governor. The governor is shown at *H* in Fig. 20, and operates a throttle valve contained in the casing *K*. A **stop valve**, by means of which the steam supply may be turned on or off to the engine, is contained in the same casing, and is operated by a hand wheel *L*. Stop valves will be fully explained in Chapter XI.

The **boiler** is shown in section in Fig. 20. It consists of an outer cylindrical shell *M*, made of plates riveted together. Another smaller cylindrical box *N* is contained within the shell and is riveted to it at the bottom edge. This forms the **fire box**, and is furnished with a grate formed of a number of **fire bars** *P* laid side by side with small spaces between to admit air from the **ash pit** *Q*. A number of tubes *R* are expanded tightly into the top of the fire box and into a tube plate *S* near the top of the shell. The hot gases from the fire pass upwards through the tubes, as shown by the arrows, giving up their heat to the surrounding water, and finally escape by the chimney *T*. The water level in the boiler stands at *U*, the space between this level and the tube plate *S* being filled with steam. Several hand holes, rendered tight by covers, enable sludge, etc., to be cleaned out of the bottom of the boiler; one of these is shown at *V*.

Steam is led from the boiler through a pipe *W* to the throttle valve and so to the steam cylinder. The exhaust steam from the engine cylinder escapes through a pipe *X* which is led into the chimney and so discharges into the atmosphere. The resulting upward blast ensures a draught which will cause the air necessary for the combustion of the coal to pass freely into the fire.

The boiler is kept supplied with water to make good that which is evaporated into steam by means of a **feed pump** *Y* (Fig. 21). This pump has a plunger *Z*, which forms part of the valve rod, and is fitted with suction and discharge valves and pipes which enable it, at each revolution of the engine, to discharge the necessary quantity of feed water into the boiler through the pipe *a* and a

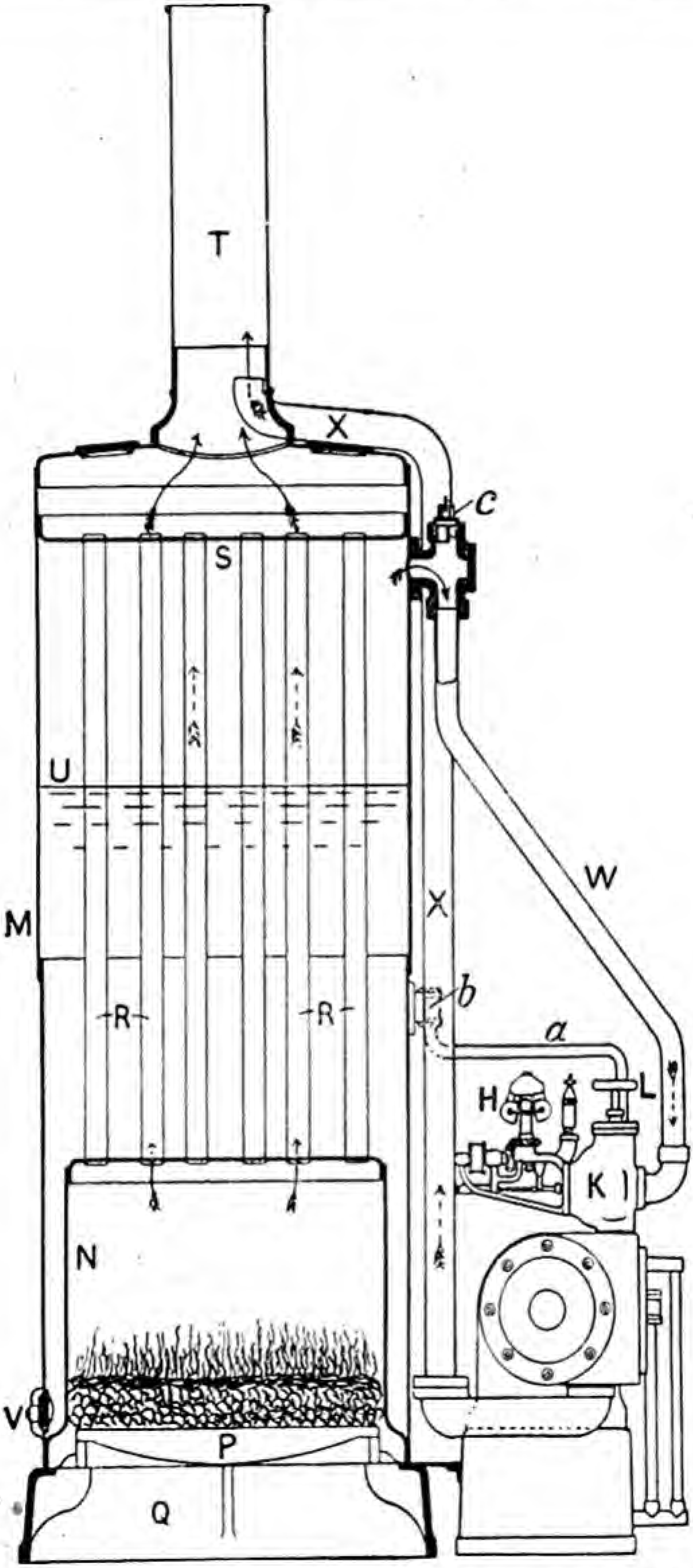


FIG. 20.—Elevation of a small horizontal steam engine, with vertical boiler; the latter is shown in section.

non-return valve *b* (Fig. 20). The **non-return valve** prevents any water flowing out of the boiler through the feed supply pipe,

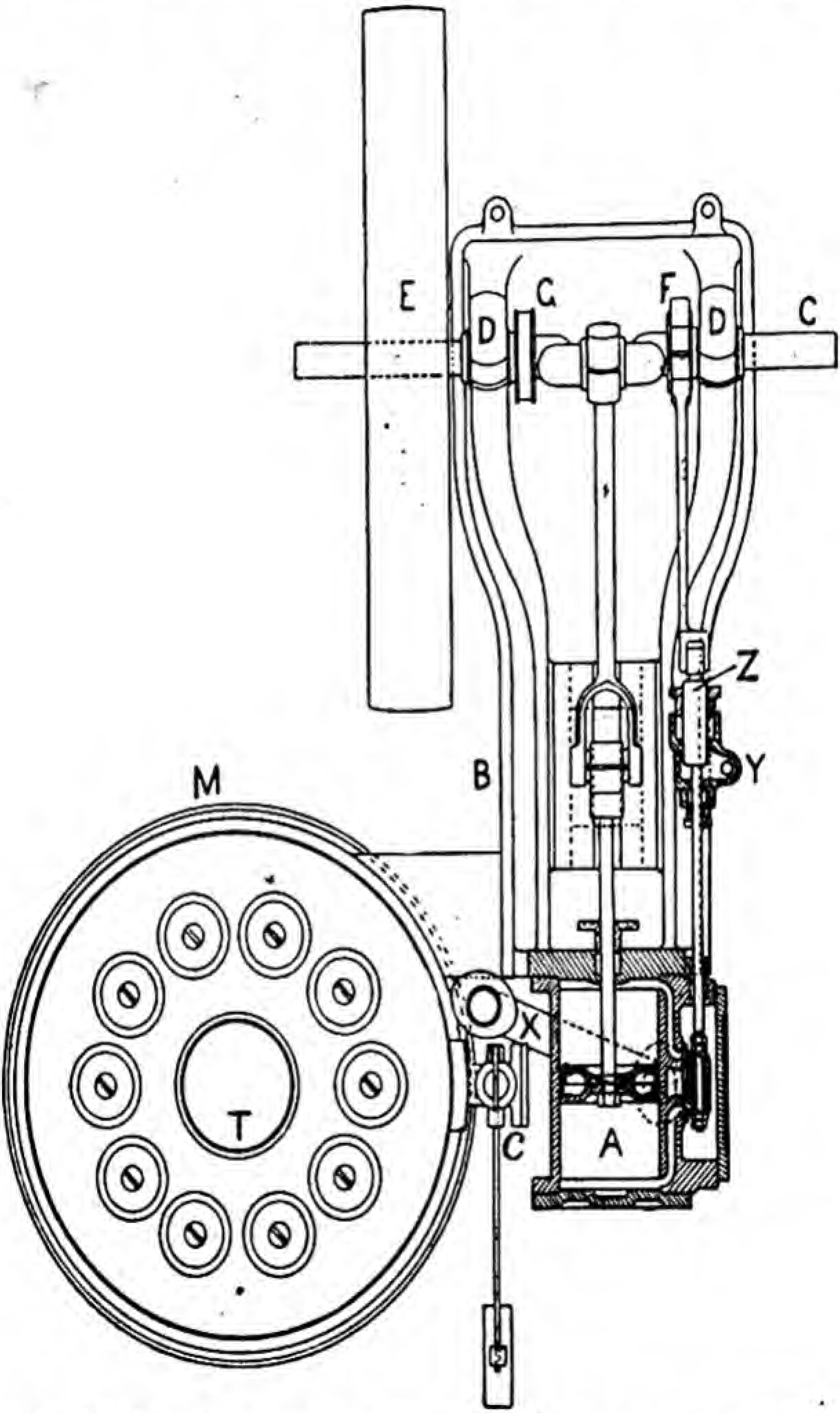


FIG. 21.—Plan of the steam plant shown in Fig. 20. The cylinder and feed pump are shown in section.

should the pump not be delivering water. The feed water arrangements will be more fully described subsequently.

A **safety valve** is fitted to the boiler at *c*; its function is to enable steam to escape from the boiler should the pressure exceed a prearranged amount and so endanger the boiler. The type of valve illustrated is called a **lever safety valve**; details of such a valve are more clearly shown in Fig. 158, where it will be seen that the valve is held down by means of a weight attached to the end of a lever which bears downwards on the valve. The resulting downward pressure on the valve is sufficient to counteract the upward steam pressure under normal working conditions, but should the steam pressure exceed the proper amount, the valve will lift and permit the escape of the steam.

The steam engine as a heat engine.—The steam engine and boiler may be looked upon as contrivances for converting into **mechanical work** the energy contained in the coal or other fuel in the form of potential **heat**. It follows, therefore, that the student who wishes to have sound notions on the subject must have clear ideas of the nature of heat. The leading elementary principles of heat are explained in succeeding Chapters. Much information which cannot be set down here will be gained by carrying out carefully the experiments described.

EXERCISES ON CHAPTER I.

NOTE TO THE STUDENT.—In giving descriptive sketches, select as far as possible types other than those given in the text. The most suitable answer will be obtained by giving sketches of a type which you have actually seen or handled.

1. Explain, with diagrams, how the motion of the piston in the cylinder is converted into motion of rotation at the shaft.

2. Give sketches showing the construction of a cylinder body for a steam engine, omitting the valve chest, but showing the cylinder covers and the piston rod stuffing-box.

3. Give sketches and describe any form of piston for a steam cylinder. Explain clearly the construction of the spring packing rings.

4. Sketch and describe a common slide valve. Explain how it is attached to the valve rod and how it allows steam to enter and discharge from the cylinder.

5. Give sketches of and describe the construction of any form of eccentric.

6. Explain how the governor controls the speed of the engine. Illustrate your answer by reference to an outline diagram.

7. Why is it necessary to guide the outer end of the piston rod? Give sketches and describe any form of crosshead and slipper.

8. Explain the construction and give sketches of the connecting rod of any engine with which you are acquainted.

9. Give sketches of the soleplate or frame of any engine you know. Show clearly how the cylinder is secured to the soleplate or frame.

10. Explain clearly the object of fitting each of the following valves to a steam boiler: (*a*) stop valve, (*b*) safety valve, (*c*) non-return feed valve.

11. Describe clearly, with sketches, the working of any single cylinder direct-acting non-condensing engine with slide valve and eccentric. Do not give too much detail, but show that you understand how the piston and stuffing-box are made steam-tight; how the piston is fastened to the rod; how the ends of the connecting-rod are made; the action of the governor and of the fly-wheel. 1897.

12. Describe, with sketches, any form of governor. 1907.

CHAPTER II.

TEMPERATURE. EXPANSION.

Temperature.—A person, on touching different bodies, may perceive that some of them are hot and others cold. The hotter bodies are said to be at a higher **temperature** than the colder ones. We may say, in fact, that **temperature means the hotness of a body as compared with some standard temperature.** Our sense of hotness often enables us to form an opinion regarding the temperature of a body, but it is not always trustworthy. Hence, some form of instrument is required for measuring temperatures, and such instruments are known as **thermometers** or temperature measurers.

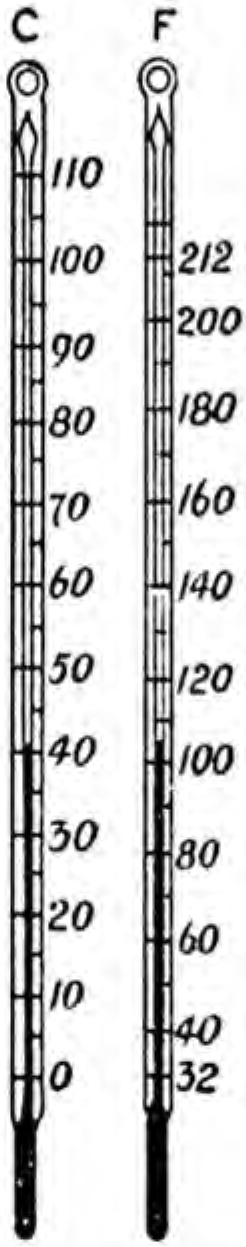


Fig. 22.—Centigrade and Fahrenheit mercury thermometers.

Thermometers.—Almost all substances expand on being warmed and contract when cooling. Advantage is taken of this property in the commonest form of thermometer, in which temperatures are measured by the amount of the expansion, or contraction, on change of temperature of mercury contained in a glass tube. The mercury thermometer consists of a fine glass tube (a capillary tube), at one end of which a bulb of either spherical or cylindrical shape is formed (Fig. 22). Mercury is introduced into the bulb and boiled to drive off any air it may contain. The other end of the tube is then sealed, so that the only contents consist as nearly as possible of mercury and vapour of mercury.

The glass walls of the bulb are blown very thin in order that the contained mercury may come quickly to the temperature of any body to which the thermometer may be applied. It will be found that the level of the mercury in the stem of the thermometer rises when the bulb is brought into contact with a hot body and falls when touching a cold body, this result being due to the expansion or contraction of the mercury in the bulb. The level of the mercury surface may be taken as an indication of the temperature of the body with which the thermometer is in contact.

Graduation of thermometer.—In order that a thermometer may be used for comparing temperatures, a **scale of temperature** must be engraved along its stem. Two **fixed points** are first marked, and the interval between them is then divided into a number of parts called **degrees**. The fixed points are:

(*a*) the level of the mercury surface when the bulb and the part of the stem containing mercury are surrounded with melting ice, this level being called **freezing point**;

(*b*) the level of the mercury surface when the bulb and the part of the stem containing mercury are surrounded with steam coming from water boiling under standard atmospheric pressure, this level being called **boiling point**. As the temperature at which water boils varies greatly with the pressure to which it is subjected, it is necessary to take a standard pressure in order to secure a definite boiling point. Standard atmospheric pressure is 760 mm., or 30 inches, of mercury, as shown by the barometer at sea-level.

Scales of temperature.—The **Fahrenheit scale** (named after its inventor) has the freezing point of water marked 32° and the boiling point 212°. The interval between these fixed points is divided into 180 degrees. Zero on this scale will be 32° below the freezing point.

The **Centigrade scale** has the freezing point of water marked 0° and the boiling point 100°; the interval between is divided into 100 degrees.

In the **Réaumur scale**, the freezing point of water is marked 0° and the boiling point 80°.

Temperatures below zero on all these scales are indicated by the negative sign. Thus −10° F. means 42 Fahrenheit degrees below the freezing point of water.

Of these scales, the first two mentioned are largely used; the

Réaumur is used only in some parts of the Continent. Engineers in this country use, for the most part, the Fahrenheit scale; physicists and chemists employ the Centigrade scale. It would be better if all were to use the latter, but in existing circumstances the engineering student must be familiar with both, and should be able to convert with facility temperatures stated on one scale into the corresponding temperatures on the other.

Conversion of temperatures.—The error which students are most liable to make in converting temperatures from the Fahrenheit to the Centigrade scale, or *vice versa*, is due to the same temperature—the freezing point of water—being marked 32° on one and 0° on the other, rendering it necessary sometimes to add and sometimes to subtract 32°. To avoid risk of error, the following method may be used. Sketch two thermometers side by side, as shown in Fig. 23. Mark one F. and the other C. Mark the fixed points on each diagram, putting corresponding points opposite one another. Mark the given temperature on the proper thermometer. Suppose this to be 60° F. It will be seen by inspection that this temperature is $(60-32)=28$ Fahrenheit degrees above freezing point. Now since 180 Fahrenheit degrees correspond with 100 Centigrade degrees, we may find the number of Centigrade degrees corresponding to 28 Fahrenheit degrees from

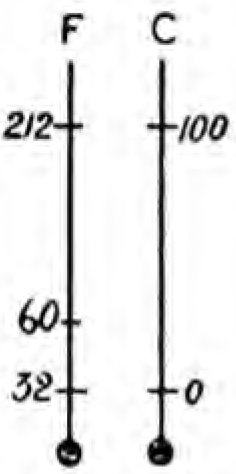


Fig. 23.—Conversion of thermometric scales.

$$\text{Cent. degrees} : 28 = 100 : 180,$$

or,

$$\text{Cent. degrees} = 28 \times \tfrac{100}{180}$$
$$= 28 \times \tfrac{5}{9} = \underline{15^\circ{\cdot}5}.$$

The given temperature, viz. 60° F., therefore corresponds with 15°·5 C.

Example i. Find the temperature on the Fahrenheit scale corresponding to 10° C.

Cent. degrees above freezing point $=10$ (Fig. 24).
Fah. " " " " $=10 \times \tfrac{180}{100}$
$= 10 \times \tfrac{9}{5} = \underline{18^\circ}.$

$\therefore$ Required temperature $= 18 + 32 = \underline{50^\circ}$ F.

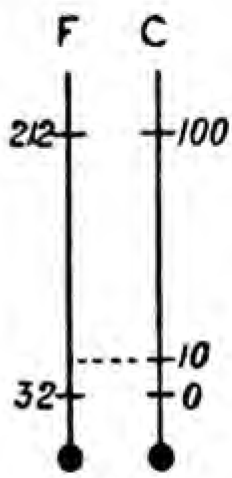


Fig. 24.

EXAMPLE ii. Find the temperature on the Centigrade scale corresponding to $-15°$ F.

Fah. degrees below freezing point $= 15 + 32 = 47$ (Fig. 25).
Cent. " " " " $= 47 \times \frac{5}{9}$
$= \underline{26°{\cdot}1}.$

$\therefore$ Required temperature $= -26°{\cdot}1$ C.

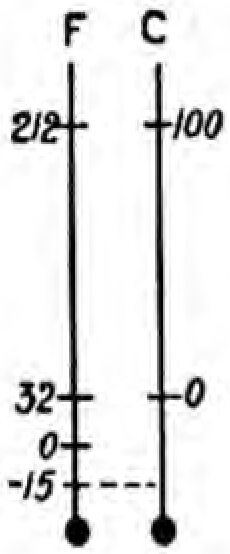


FIG. 25.

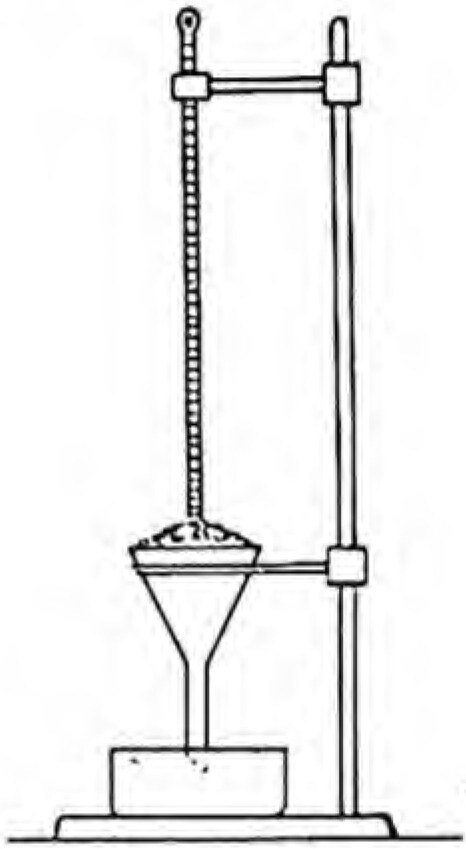


FIG. 26.—Apparatus for determining the freezing point of a thermometer.

Testing thermometers.—The following experiments should be performed carefully.

EXPT. 1.—Arrange a funnel and beaker on a retort stand, as shown in Fig. 26. With a chisel remove some shavings from a block of ice, and put them into the funnel. Insert a thermometer, and pack the ice shavings closely round the bulb and stem as far up as the freezing point graduation. Bring the eye to the same level as the top of the mercury column, and take readings at intervals. Note the final steady reading; this may be taken as the true freezing point of water. The freezing point error of the instrument will be the difference between the final steady reading and 32° if a Fahrenheit thermometer has been used, or 0° if it is a Centigrade thermometer. The error should be noted with its proper sign, + or −, attached.

Observe in carrying out this experiment that the temperature as shown by the thermometer remains steady during the whole time that the ice is melting.

EXPT. 2.—Bring some water to boiling temperature in a flask fitted with a side branch through which the steam evolved may be discharged (Fig. 27). By means of a cork fit a thermometer to the mouth of the flask. Notice that the temperature remains steady at or near the boiling point graduation during the whole time that the water is boiling.

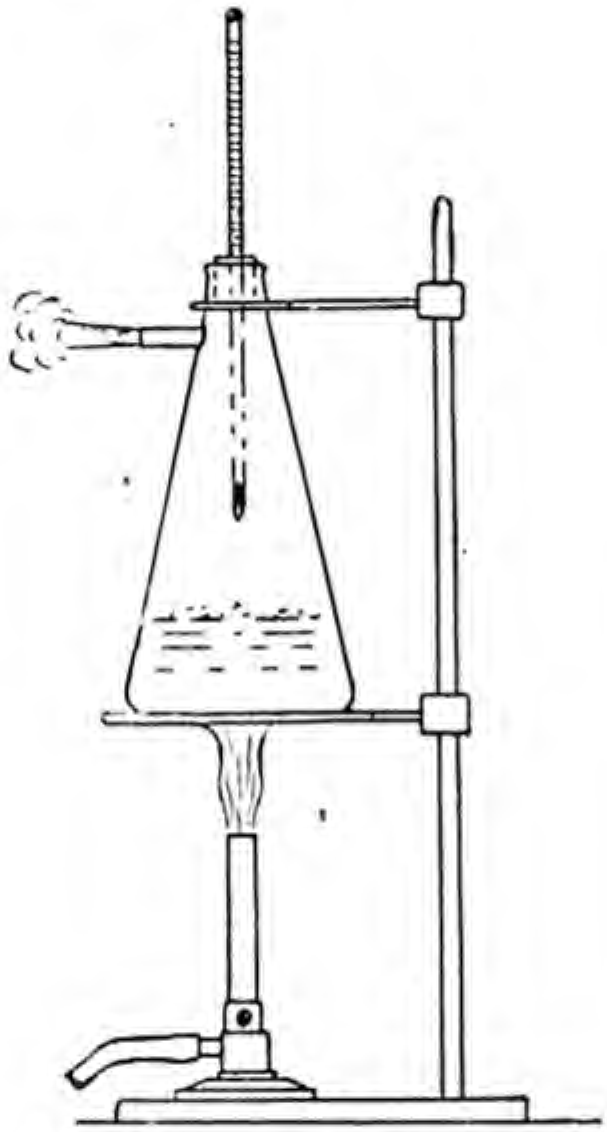


FIG. 27.—Simple apparatus for determining the boiling point of a thermometer.

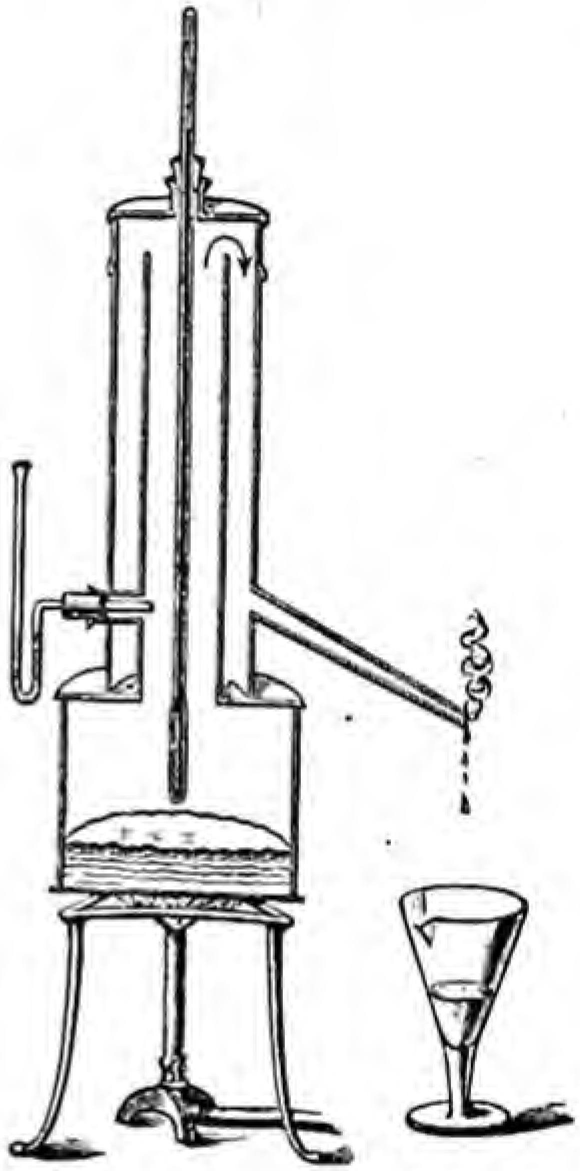


FIG. 28.—Apparatus for determining the boiling point of a thermometer.

A better form of apparatus for this experiment is shown in Fig. 28. This consists of a small copper boiler, to the cover of which a double copper tube is attached. The thermometer under test is placed in the inner tube, and is surrounded by steam coming from the boiling water. This steam passes up the inner tube, then down the outer tube, being finally discharged at the bottom. The object of this arrangement is to **steam-jacket** the tube containing the thermometer, thereby ensuring that the tube shall be at the same temperature as the steam. A small glass U-gauge containing water is connected to the outer tube. When the water stands at the same level in both limbs of this gauge, the pressure of steam inside the apparatus is equal to

that of the atmosphere outside. In using this apparatus, the thermometer is pushed through a cork, and placed in position so that its boiling point graduation is just visible above the cork. The water is then brought to boiling, and after steam has been given off freely during a few minutes, readings of the thermometer are taken.

Should the reading in this experiment differ from the boiling point graduation of the thermometer, it does not necessarily follow that the thermometer is in error. It will be remembered (p. 19) that the water must be boiling under standard atmospheric pressure, and this may not be the pressure of the atmosphere at the time the test is carried out. To obtain the pressure of the atmosphere, readings of a standard barometer should be taken while the experiment is going on, and the temperature at which water boils when subjected to this pressure will then be found from the Table, p. 456. The error may be stated approximately as the difference between the observed boiling temperature and that shown in the Table, and should be noted with its proper sign attached, + or −.

Should the bore of the stem of a thermometer not be uniform, equal increases of volume of the mercury will not be indicated by equal differences in the level of the mercury column. The bore, of course, ought to be as uniform as possible, and this is tested in ordinary physical work by careful measurement, and any inequalities found allowed for. For our purpose, the following simple experiment will suffice for testing the accuracy of the graduations between the freezing and boiling points.

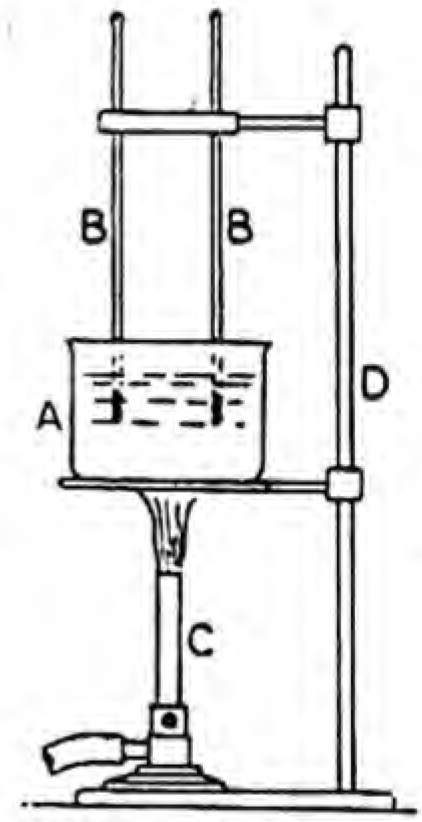


FIG. 29.—Apparatus for comparing the scale of a thermometer with that of a standard thermometer.

EXPT. 3.—In Fig. 29, *BB* are two thermometers, one of which is a standard thermometer, *i.e.* one in which the graduation errors are known, and the other is a thermometer to be tested. Both are suspended with their bulbs immersed in water contained in a beaker *A*. Gradually raise the temperature of the water, and take simultaneous readings of the thermometers at intervals of say 5°, being careful to stir

the water well before taking the readings. Note these readings thus:

Standard thermometer.		Thermometer under test.	
Observed temp.	True temp.	Observed temp.	Error.

Columns 1 and 3 are filled in from the observations; column 2 from the known errors of the standard thermometer; column 4, obtained by taking the differences of columns 2 and 3, shows the errors of the thermometer under test at various parts of the scale.

Care and use of thermometers.—Thermometers should be handled carefully; no attempt should be made to force the thin-walled bulbs through corks; thermometers are liable to be injured if subjected suddenly to great changes of temperature. Do not use any thermometer in a place where there is risk of its being subjected to a temperature higher than that to which it is graduated, otherwise the bulb may be burst by the pressure of the expanding mercury. This danger may be guarded against partially by using thermometers having a safety bulb blown at the top of the stem (Fig. 22).

Fig. 30.—Hopkinson's pressure cup for thermometers.

In obtaining the temperature of water or steam under pressure, a metal cup closed at its inner end may be secured to the containing vessel (Fig. 30). Some cylinder oil, or mercury, is poured into the cup and comes quickly to the temperature of the contents of the vessel, and this may be measured by inserting a thermometer in the liquid. This arrangement prevents any risk of the thermometer bulb being collapsed by the pressure inside the vessel.

In many cases where differences in temperature at two parts of a pipe are required, it suffices to secure the thermometer with its stem lying along the pipe, and to wrap cotton waste or flannel round the pipe over the bulb, so as to ensure that the mercury comes to the same temperature, approximately, as the pipe. As both thermometers are under similar conditions, the difference in their readings will very nearly equal the difference in the temperatures of the contents of the pipe at the two places.

Measurement of high temperatures.—Under ordinary atmospheric pressure, mercury boils at 357° C, consequently ordinary mercury thermometers cannot be used for measuring temperatures higher than this. High temperatures may sometimes be stated with sufficient exactness by reference to the known melting temperatures of certain substances. Thus, we may say that the temperature of a body is about that of melting lead (617° F.), if the temperature be such that a small piece of lead placed in contact with the body just melts. Paraffin, sulphur, tin may be used in the same way. The method is useful for roughly determining the temperature of furnaces. Substances used in this way are called **thermoscopes.**

Pyrometers are instruments used for determining high temperatures with considerable accuracy. The temperature of a flue or furnace may be found by inserting a piece of platinum, copper, or other substance, allowing it to remain some time so as to come to the temperature of the furnace, then removing and plunging it into water. From the known weights of the substance and of the water used, and the temperature of the water before and after, the temperature of the flue or furnace may be calculated by a method which will be explained later.

The electrical resistance of platinum wire at varying temperatures is used in some pyrometers to indicate the temperature to which the wire is exposed. In other pyrometers advantage is taken of the varying strength of electric current set up in an outer circuit when two dissimilar metals, such as platinum and iridium, in contact with one another, and in the circuit, are exposed to different temperatures. Other electrical and optical methods are in use, but the details of all of them are beyond the scope of this book, although their use in practice has been

rendered exceedingly simple in modern instruments of appropriate design.

Expansion.—The expansion of metals as a consequence of elevation of temperature is well known to engineers. Advantage is taken of the property in the execution of work requiring **shrinking**. One piece of material, such as a wheel tire, or a gun tube, has to fit tightly on to another piece. The outer piece is bored out too small to go on when cold; but on being heated, it expands, and may then be slid into place. As cooling goes on,

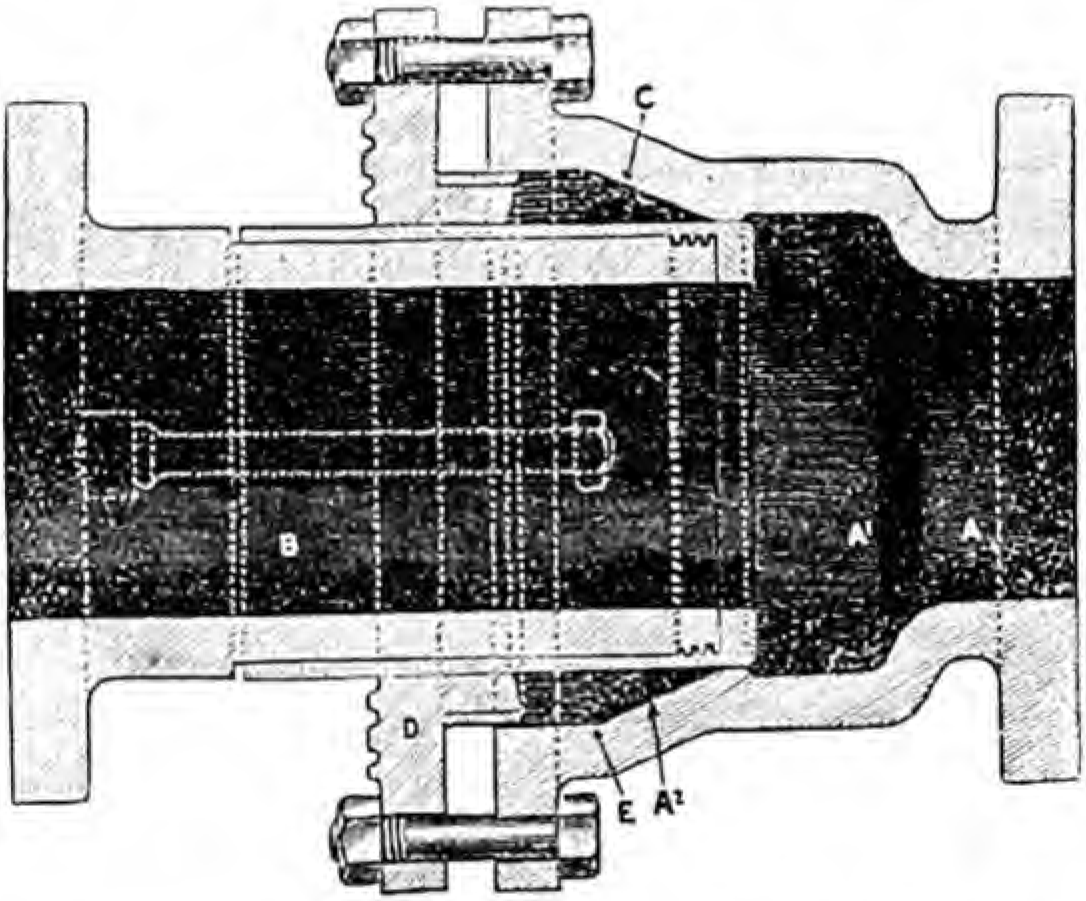


FIG. 31.—Hopkinson's arrangement for permitting a line of pipes to expand on elevation of temperature.

the outer piece shrinks, and binds itself tightly on to the inner piece of material.

In other classes of work, such as rails, steam pipes, boiler furnace tubes, etc., the expansion due to heating is a nuisance which has to be provided for.

Fig. 31 shows an arrangement for giving freedom to expand to steam pipes. In this arrangement, the end B of one part of the pipe may slide inside the other part A, which is formed at A' to receive it. The joint is made by means of packing E, A_2, inserted in the circular box C, and forced down tight by the gland D. Two long studs, one shown dotted, are screwed

into the flange of B, pass through holes in the flange of A, and prevent any danger of the pipes becoming separated by the internal pressure.

Expt. 4.—Apparatus by means of which an experiment useful for showing the expansion of a metal tube when heated is shown in Fig. 32. A is a small boiler connected by rubber tubing B to a copper tube C. The copper tube is about 3 feet long and is plugged at both ends; a branch is soldered near each end on opposite sides of the tube. Steam from the boiler enters the tube through B and is discharged freely through K. Two brass plates, each about $3\frac{1}{2}''$ long, 1" wide, and $\frac{1}{4}''$ thick are soldered to the tube at D and F. The tube is supported

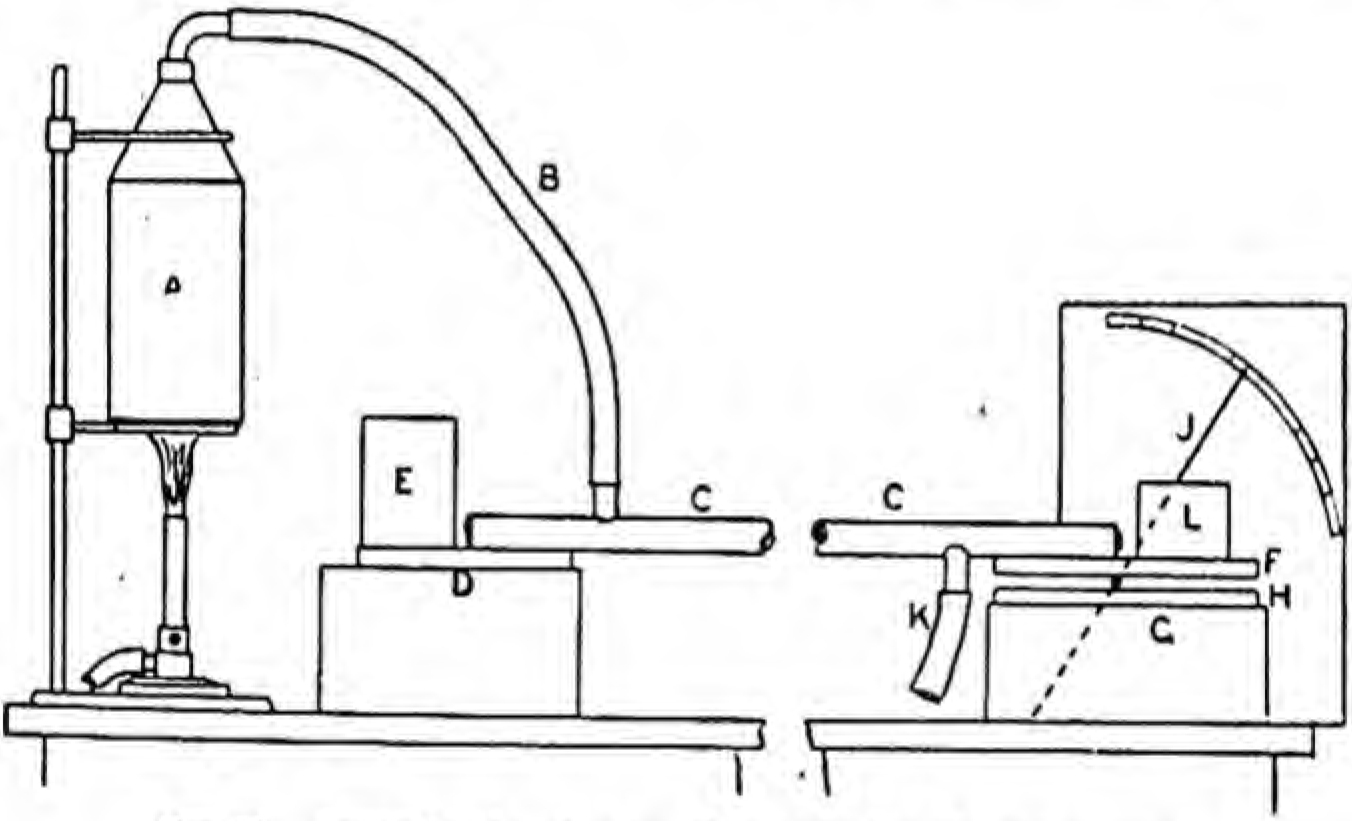


Fig. 32.—Apparatus for showing the expansion of a metal tube.

by a block at D and held down by a weight E placed on the brass plate. The brass plate at the other end of the tube rests on a small roller made of thin drill steel which may roll on a brass plate H secured to a supporting block G. Any movement horizontally of this end of the copper tube will cause the roller to rotate, and this rotation will be rendered evident by the pointer J travelling over the graduated scale. The pointer may be made of a narrow strip of card secured to the roller by means of sealing-wax. On passing steam from the boiler through the copper tube, the expansion will be shown clearly by the movements of the pointer.

Expt. 5.—The expansion of water when heated may be shown by using a small flask (Fig. 33) fitted with a rubber stopper and having a glass tube inserted. A paper scale is secured to the glass tube. Water

is introduced to the flask and stands, when cold, two or three inches up the tube. The water is best coloured with some red ink. On placing the flask into a vessel containing hot water, the water in the flask will be warmed gradually and its consequent expansion will be shown by the level rising in the glass tube.

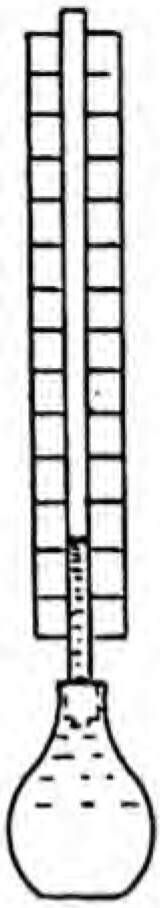


FIG. 33.—Apparatus for showing the expansion of water.

Coefficient of linear expansion.—All metals do not expand to the same extent on being heated through the same range of temperature. Thus, copper and brass expand more than iron for the same increase of temperature. This fact may be illustrated by means of two flat bars, one of copper, the other of iron, riveted together and heated. If straight to begin with, the composite bar will be found to be bent after heating, the copper bar being on the convex side of the bend, showing that it has expanded more than the iron.

The coefficient of linear expansion of a substance is the increase in length which a bar of unit length undergoes when heated through one degree.

Let e = coefficient of linear expansion,
L = original length of bar,
t = elevation of temperature.

Then, the increase in length of a bar of unit length, heated through t degrees $= t \times e$; and for a bar of length L, the increase in length $= L \times t \times e$;

$$\therefore \text{ length of bar after heating} = L + Lte = L(1 + te)$$

EXAMPLE. Steel rails, each 20 feet in length, are laid when the atmospheric temperature is 50° F. It is intended that the ends should touch if the temperature reaches 120° F. What space should be left between the ends when laying the rails? Take the coefficient of linear expansion $= 0{\cdot}0000067$ per degree Fah.

$$\text{Increase in temperature} = (120 - 50) = 70^\circ \text{ F.}$$

$$\begin{aligned}\text{Increase in a length of 20 feet} &= Lte\\ &= 20 \times 70 \times 0{\cdot}0000067\\ &= 0{\cdot}00938 \text{ foot}\\ &= 0{\cdot}113 \text{ inch.}\end{aligned}$$

The space left between the ends in order to allow for this increase in length must be 0·113 inch.

Coefficient of superficial expansion.—This coefficient may be defined as the **increase in area which a plate of unit area undergoes when heated through one degree.**

The value of this coefficient for a given substance may be shown to be **double that of the linear coefficient of expansion** for the same substance.

The **coefficient of cubical expansion** of a substance may be defined as the **increase in volume which unit volume will undergo when heated through one degree.**

The value of this coefficient may be shown to be **three times that of the linear coefficient of expansion of the substance.**

In using the numerical values of the above coefficients, care should be taken to ascertain whether the values given in the Table from which they are extracted are for the Fahrenheit or Centigrade scale. In books on engineering subjects they are generally stated for the Fahrenheit scale (such a Table will be found on p. 456), and in physical text books, for the Centigrade scale.

EXERCISES ON CHAPTER II.

1. Explain the terms "temperature," "scale of temperature," "freezing point," "boiling point."

2. Find the temperature Fahrenheit corresponding to 140° Centigrade.

3. Find the temperature Centigrade corresponding to −40° Fahrenheit.

4. Find the temperature Fahrenheit corresponding to −273° Centigrade.

5. A bar of brass measures 34″ in length at a temperature of 60° Fahrenheit. What will be its length at a temperature of 200° Fahrenheit? Take coefficient of linear expansion = 0·0000105 per degree Fah.

6. A crank has to be shrunk into its position on the shaft. The hole in it is 12·02″ in diameter at 60° F. Calculate to what temperature it must be raised in order that the diameter of the hole may be 12·05″. Take coefficient of linear expansion = 0·0000067 per degree Fah.

7. Give sketches and description of any means for providing for the expansion of a long line of steam pipes.

8. Calculate the change in length of a line of wrought iron steam pipes, 65 feet long, when the temperature is raised from 50° F. to 338° F. Take coefficient of linear expansion = 0·0000067 per degree Fah.

CHAPTER III.

HEAT AND ITS MEASUREMENT.

Quantity of heat.--When a hot and a cold body are brought into contact, both will come ultimately to the same temperature, which will lie between the temperatures originally possessed by the bodies.

EXPT. 6.—Take two vessels, one, *A*, containing about 2 pints of water at about 60° F., the other, *B*, containing about ½ pint of water at about 150° F. Place a thermometer in each vessel, stir the water and take the temperatures. Now pour the water from *B* into *A*, stir again and read the temperature. With the quantities as mentioned above this will probably be about 70° F. As the water originally in *A* has been warmed about 10° F., while that in *B* has been cooled about 80° F., it is evident that something has passed from the hotter into the colder water. To this something is given the name of heat.

Notice that temperature and heat are not the same. As the temperature of the water in *A* has not been increased to the same extent that the temperature of the water in *B* has been lowered, it is evident that it has not been temperature that has been transferred. Moreover, a comparatively cold body may contain a great quantity of heat while a very hot body may possess a small quantity of heat only. A vessel of water set to boil over a Bunsen burner receives a great quantity of heat, as estimated by the time taken, while its final temperature of 212° F. is comparatively low. A wire held in a flame comes to a very high temperature almost immediately, and evidently can only contain a small quantity of heat.

Units of heat.—Quantities of heat are measured by comparison with that quantity required to elevate the temperature of unit mass of water through 1°. In the metric system, the **gram-degree-**

Centigrade is the unit of heat; this unit is **the quantity of heat required to raise the temperature of one gram of water through 1° C.** This unit is sometimes called a **therm**, or a **gram-calorie.** Frequently a unit of heat 1000 times as large as the therm is used, this being called a **calorie**, or sometimes a **major-calorie** or **great calorie.**

In Britain, two heat units are in common use as well as the metric units. These are:

(*a*) **the quantity of heat required to raise the temperature of 1 lb. of water through 1° C.**;

(*b*) **the quantity of heat required to raise the temperature of 1 lb. of water through 1° F.** The former of these is called the **pound-degree-Centigrade unit**, or, briefly, the **Centigrade unit** of heat; the latter is called the **pound-degree-Fahrenheit unit**, or the **Fahrenheit unit**, or, more generally, the **British Thermal Unit**, written B.T.U.

Engineers use most frequently the British thermal unit, although recently in several important papers on engineering subjects the pound-degree-Centigrade unit has been employed. Students should make themselves familiar with all three units.

Example i. Calculate the quantity of heat in British thermal units required to raise the temperature of one pound of water through 1° C.

Since 1 B.T.U. can raise the temperature of 1 lb. of water through 1° F., and since $\frac{9}{5}$ degree Fah. is equivalent to 1 degree Centigrade, it follows that $\frac{9}{5}$ B.T.U. will be required.

From this example it will be seen that

1 lb.-degree-Cent. unit $= \frac{9}{5}$ B.T.U.

1 B.T.U. $= \frac{5}{9}$ lb.-degree-Cent. unit.

These values enable us to convert from one system to the other.

Example ii. What factor must be employed to convert a given quantity of heat stated in gram-calories into B.T.U.?

1 gram-calorie can raise the temp. of 1 gram water through 1° C.
453·6 gram-calories ,, ,, ,, 453·6 grams ,, ,, 1° C.

And, since 1 lb. = 453·6 grams, this statement may be written,

453·6 gram-calories can raise the temp. of 1 lb. water through 1° C.
$(\frac{5}{9} \times 453{\cdot}6)$,, ,, ,, ,, ,, ,, 1° F.

$$\therefore\ 1\ \text{B.T.U.} = \tfrac{5}{9} \times 453{\cdot}6$$
$$= \underline{252}\ \text{gram-calories.}$$

Hence, to convert from gram-calories to B.T.U. multiply the gram-calories by $\frac{1}{252} = 0{\cdot}003968$.

To convert from B.T.U. to gram-calories, multiply the B.T.U.'s by 252.

Specific heat.—Suppose 1 lb. each of several different substances to be experimented on with a view to ascertain what quantity of heat is required to raise the temperature of each through 1°, it would, in general, be found that the quantities required differ. Water requires most, metals require a much smaller quantity of heat. The **specific heat** of a substance may be defined as **the quantity of heat required to raise the temperature of unit mass of the substance through one degree.** The specific heat of water by this definition will be 1. The specific heat of iron is about $\frac{1}{9}$, from which we infer that $\frac{1}{9}$ B.T.U. when imparted to 1 lb. of iron will raise its temperature through 1° F. It will also be seen that since 1 lb.-degree-Cent. unit can raise the temperature of 1 lb. of water through 1° C., $\frac{1}{9}$ lb.-degree-Cent. unit will raise the temperature of 1 lb. of iron through 1° C. In the same way, $\frac{1}{9}$ gram-calorie unit can raise the temperature of 1 gram of iron through 1° C. It will therefore be understood that the number giving the specific heat of a substance is the same for all three systems of units.

A Table giving the specific heats of some common substances will be found on p. 457.

Water equivalent of a body.—By this expression is meant that weight of water which requires the same quantity of heat to raise its temperature through one degree as the body would require in order to elevate its temperature to the same extent. Thus, the water equivalent of a piece of iron weighing 1 lb. is $\frac{1}{9}$ lb.; and of an iron cylinder weighing 2 cwts. is $\frac{224}{9}=25$ lbs. water nearly. This conception is useful in simplifying the calculations required in reducing data obtained in experiments on transference of heat.

Let W = the weight, in lbs., of a body,
s = the specific heat of the material.

Then, Water equivalent of the body $= Ws$ lbs.

Transference of heat.—Heat may be transferred from one body to another by (*a*) **conduction,** (*b*) **convection,** (*c*) **radiation.**

In **conduction,** the particles of a body nearest to a source of heat are warmed first; these then heat the neighbouring particles, which in turn pass on heat to the others; heat is thus transferred through chains of particles until the whole body has been heated.

In **convection**, particles nearest to the source of heat are warmed and then pass off to some other part of the body, making room for other particles to approach the source of heat to be warmed and pass off in turn. The whole body ultimately becomes warmed by these currents of particles.

In **radiation**, heat is transferred by a kind of wave motion in the ether, a medium which, it is assumed, fills all space. The heat waves arriving at a body and being absorbed produce the ordinary effects of heat.

All these methods of transference of heat occur in steam boilers. The furnace plates become heated by heat radiated from the fire; heat passes through the plates into the water by conduction; and the water becomes heated by convection, the resulting currents setting the water into circulation.

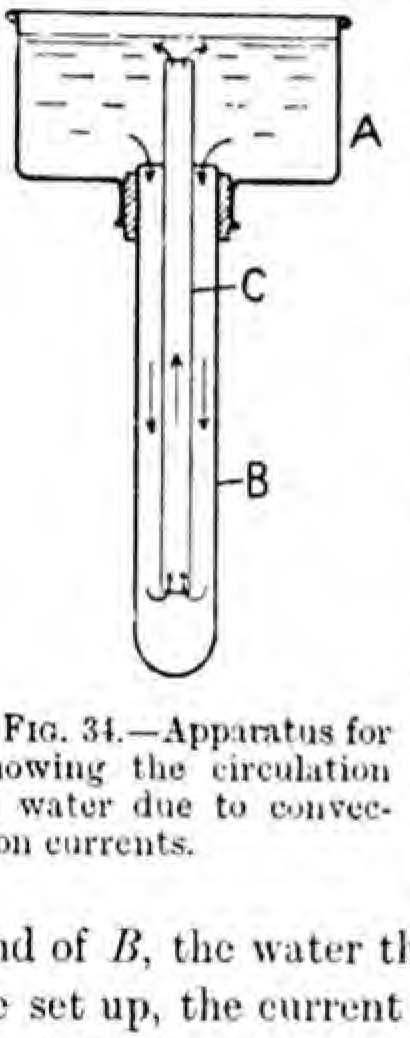


FIG. 34.—Apparatus for showing the circulation of water due to convection currents.

EXPT. 7.—Fig. 34 shows an apparatus by means of which convection currents in water may be studied. *A* is a vessel with a hole in its bottom made to receive a glass tube *B*. *B* is sealed at its lower end, and its upper end is about flush with the bottom of *A*. Another smaller bore glass tube *C*, open at both ends, is suspended centrally in *B*, its lower end being about 2″ from the bottom of *B*, and its upper end being a little below the surface of water which is contained by *A* and fills both tubes. On applying a Bunsen flame gently at the lower end of *B*, the water there will be warmed and convection currents will be set up, the current flowing up through *C* to the surface of the water in *A*, while colder water descends from *A*, through the space between the two tubes, to be heated in turn.

This device is occasionally used in steam boilers. The importance of making proper provision for free circulation of the water in steam boilers should be here noted.

Calculation regarding heat transference.—In making calculations of the ultimate temperature attained when heat is transferred from one body to another, we may assume in the first instance that no heat is wasted in raising the temperature of any body other than the colder one considered. Corrections may then be

estimated and applied for any heat known to be wasted. Calling the two bodies A and B, we may state as an approximate solution:

Heat passing from A = heat entering B.

EXPT. 8.—Weigh about 4 lbs. of cold water in a vessel, preferably made of copper, and about 1 lb. of hot water in another vessel. Take the temperatures, using two thermometers, one in each vessel. Then pour the hot water into the first vessel, stir up well, and note the final steady temperature.

Compare this temperature with the value calculated as follows:

Let W_A = weight in lbs. of the cold water.
W_B = " " " " hot "
$t°_A$ = temperature of the cold water.
$t°_B$ = " " hot "
$t°$ = calculated final steady temperature.

The temperatures are to be stated all in the same scale.

Heat passing from $B = W_B \times$ fall in temperature from $t°_B$ to $t°$.

Heat entering $A = W_A \times$ rise in temperature from $t°_A$ to $t°$.

Assume

Heat passing from B = heat entering A.

$$W_B(t_B - t) = W_A(t - t_A) \qquad (1)$$

$$W_B t_B - W_B t = W_A t - W_A t_A,$$

$$W_B t_B + W_A t_A = W_A t + W_B t = t(W_A + W_B),$$

$$t = \frac{W_A t_A + W_B t_B}{W_A + W_B} \qquad (2)$$

Inserting the experimental values on the right hand side of this equation, the calculated value of t will be found. This value will be found to be somewhat higher than that found for t experimentally, the difference representing the total correction which must be applied to the calculated value in order to obtain the true one.

Setting aside errors in measuring the temperatures accurately, and the possible error due to some of the hot water being left in its original vessel, the principal source of error lies in the fact that the vessel in which the mixture is effected has its temperature raised as well as the water which it contains. To allow for this, the water equivalent of the vessel should be calculated and added to the weight of cold water taken.

Let W = weight of vessel,

s = specific heat of its material.

Then, Ws = water equivalent of vessel.

Equation (1) above now becomes

$$W_B(t_B - t) = (W_A + Ws)(t - t_A) \quad \ldots\ldots\ldots\ldots (3)$$

giving

$$t = \frac{(W_A + Ws)t_A + W_B t_B}{W_A + Ws + W_B} \quad \ldots\ldots\ldots\ldots (4)$$

Try if the final temperature as calculated from equation (4) agrees more nearly with that found experimentally.

Vessels, such as that used in Expt. 8, in which the heat transferences take place, are called **calorimeters.** We may define a calorimeter as **an instrument used for measuring quantities of heat.**

Expt. 9.—The specific heat of a solid may be determined by a simple method of mixtures. By means of a piece of cotton suspend a piece of iron, copper, brass, or other metal in a beaker of boiling water. Keep the water boiling for several minutes. Have ready a copper calorimeter containing a quantity of water, and adjust the temperature of this water so that it is only slightly below that of the room. This may be effected by adding hot or cold water as required. When ready, note the temperature t_1 of the water in the calorimeter; rapidly transfer the metal from the beaker to the calorimeter, taking as little boiling water as possible in doing so, and keep it moving about in the calorimeter. Note the highest temperature attained by the water in the calorimeter, calling this t_2. Remove the metal and weigh the calorimeter and water, deduct from this the weight of the calorimeter, the result being the weight of the water alone. Weigh the metal.

Let W = weight of metal.

W_w = ,, ,, water.

$W_c s_c$ = water equivalent of the calorimeter.

$212°$ = initial temp. of metal,

t_1 = ,, ,, ,, water,

t_2 = final ,, ,, ,, and metal.

(all on Fah. scale.)

s = specific heat of metal under test.

Heat passing from metal = W × drop in temp. of metal × s.

Heat entering calorimeter

$$=(W_w + W_c s_c) \times \text{rise in temp. of calorimeter.}$$

$$\therefore \quad W(212 - t_2)s = (W_w + W_c s_c)(t_2 - t_1),$$

$$s = \frac{(W_w + W_c s_c)(t_2 - t_1)}{W(212 - t_2)}.$$

From which s may be found.

In carrying out Expt. 9, if the initial temperature of the water in the calorimeter be above that of the air in the room, heat will pass into the atmosphere from the calorimeter throughout the experiment. On the other hand, if the water temperature be initially considerably below that of the room, heat will pass from the atmosphere into the calorimeter throughout the test. In either case, a correction must be applied. The necessity for this correction disappears if the initial temperature of the water is as much below the temperature of the room as the final temperature is above it. With a little trouble in adjusting the initial temperature, these conditions may be approximately secured.

Expt. 10.—Procure a number of half-gallon tin cans, without handles, and fitted with lids. Have a short piece of brass tube about $\frac{3}{4}''$ diam. soldered centrally to each lid; this will permit of a thermometer being inserted through a cork fitted to the tube. Leave one can with its surface bright; coat the whole of the outside of a second with lamp black; cover another, lid included, with cotton wool; another with hair felt; another with asbestos or any other boiler or pipe covering composition available. Set these prepared cans in a row on the bench and pour equal quantities of hot water into each through the tube, using a glass funnel and being careful that no water is spilled over the outside of the covering material. Insert thermometers and read their temperatures at intervals of 5 minutes. Tabulate thus:

Time.	Temperatures of water in				
	Can A bright tin.	Can B lamp black.	Can C cotton wool.	Can D hair felt.	Can E asbestos.

Plot these temperature readings and times on a single sheet of squared paper. The plotted curves will enable us to infer the rate at which heat passes from each can by conduction through the material and by radiation from the surface. A rough comparison of the value of each of the coverings used as a non-conductor of heat may now be made. Pay special attention to cans A and B. Notice that can B loses heat more rapidly than can A, from which we may infer that if we are compelled to use a bare metal surface for a vessel containing a substance, the temperature of which is to be preserved as nearly constant as possible, the surface should be brightly polished.

Some important definitions.—**Work** is said to be done when a force acts through a distance, overcoming resistance. Thus, work is done against the resistance of gravity while a weight is being raised. The **unit of work** generally employed is the foot-lb., expended when a force of one pound acts through a distance of one foot. **Energy** means capability of doing work. For example, a weight, when raised, possesses energy, because it can perform work if allowed to descend. All bodies in motion possess energy, for work may be performed by them while they are coming to rest. This statement evidently is true no matter how small the body may be.

Energy has many different forms. That possessed by a raised weight or coiled spring is said to be **Potential Energy**. That possessed by a moving body by virtue of its motion is called **Kinetic Energy**. Potential energy may be converted into kinetic energy ; for example, a raised weight, if allowed to fall freely, will have its potential energy gradually converted into kinetic energy during the fall, the conversion being complete at the instant the weight reaches the ground, when all the potential energy will have been converted into an equal quantity of kinetic energy. The principle of the **conservation of energy** asserts that **energy cannot be created or destroyed**, it can only be converted from one form into another.

Heat is a form of energy.—At one time it was supposed that heat was a material substance, capable of being soaked in or squeezed out, as it were, by a body. Rumford showed by his cannon-boring experiment, in which a blunt boring tool was used, that sufficient heat was evolved to boil a large quantity of water while only a very small quantity of material was removed by the

tool. He concluded it was impossible that the large quantity of heat given to the water could have been contained by and squeezed out from the small amount of material removed, and that therefore heat must be a form of motion.

Davy further confirmed this view by rubbing two blocks of ice together, taking precautions to ensure that no heat could be communicated to them from outside sources. He found that in a short time the ice was melted. Apparently an unlimited quantity of heat can be produced by the simple process of

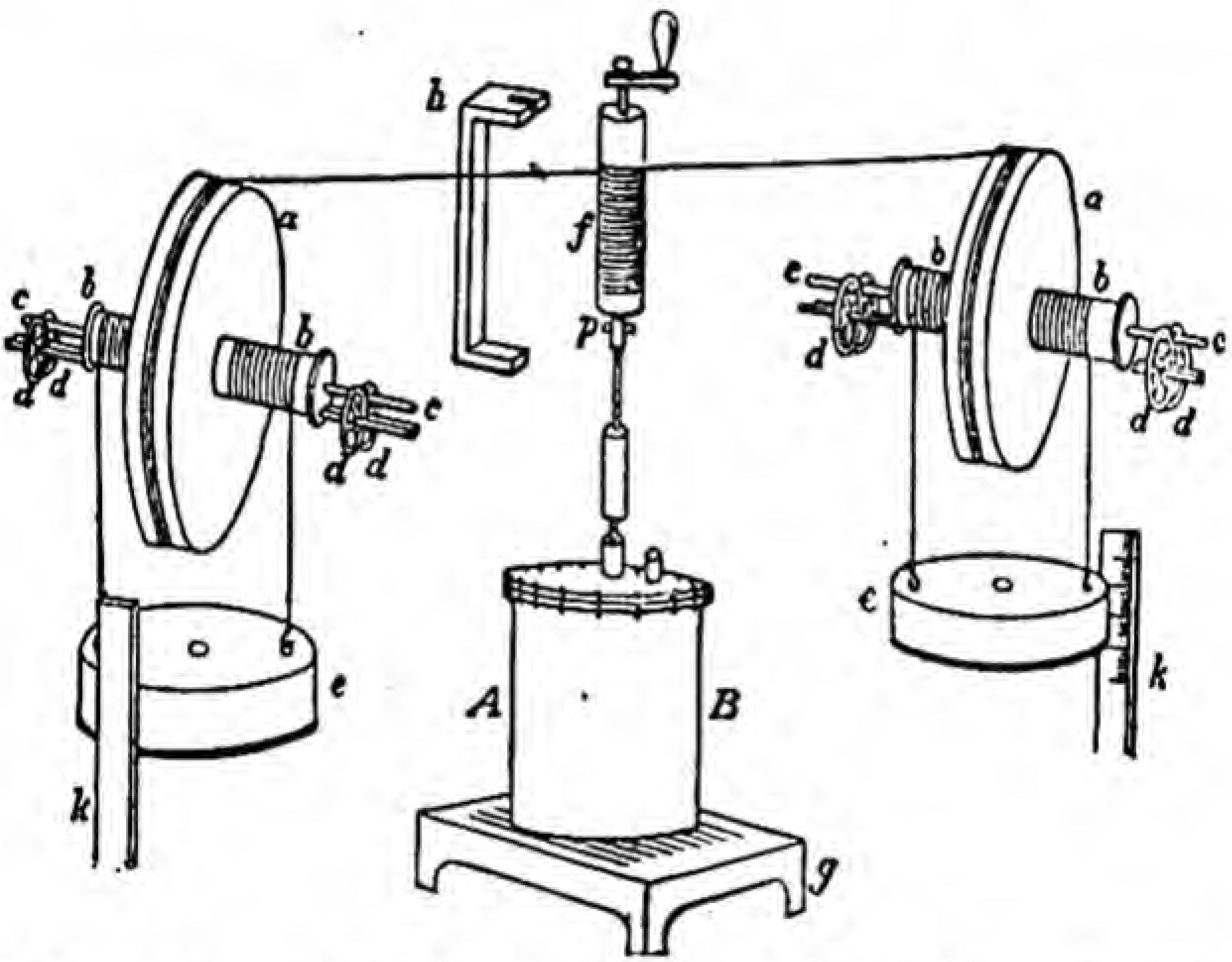


FIG. 35.—Apparatus used by Joule in his experiments on the mechanical equivalent of heat.

rubbing two bodies together, and therefore it is impossible that heat can be a material substance contained by the rubbing bodies.

Nature of heat.—It is now believed that heat is really the energy of the molecules of which any body is constructed. The molecules in a solid do not move about inside the body, that is, do not alter their relative positions, but are in a state of vibration. Heat imparted to a solid increases the molecular vibrations and so increases the energy of the molecules.

In a liquid, the molecules are not only in vibration, but may also move relatively to one another with comparative freedom.

Heat imparted to a liquid may increase the energy of vibration and at the same time produce currents of molecules from one part of the liquid to another.

FIG. 36.—Joule's calorimeter.

In substances in the gaseous state, the molecules are in rapid motion, continually colliding with one another and with the walls of the containing vessel. This continual bombardment produces pressure on the walls. Heat imparted to the gas increases the speed of the molecules, thereby increasing their kinetic energy and the pressure on the walls of the vessel.

Joule's mechanical equivalent.—Joule investigated the question of how much mechanical work must be done for the production of a given quantity of heat. In his experiments, falling weights *ee* (Fig. 35) were used to drive a paddle revolving inside a vessel *AB* fitted with baffle plates and containing water. The vessel is shown separately in Fig. 36. The work done by gravity on the falling weights was thus converted into heat by stirring the water against the resistance offered by the baffle plates. From the known weights and the height fallen, the mechanical work done was estimated, due allowance being made for mechanical losses. The rise in temperature of the measured quantity of water in the vessel enabled the heat produced to be calculated, corrections for wasted heat being applied. The result of these very careful experiments was that 772 foot-pounds of mechanical energy disappear in the production of one British Thermal Unit.

Later experiments by Rowland, Osborne Reynolds and Griffiths

give 774 and 778 as more correct numbers. As the difference between 772 and 778 does not amount to 1 per cent., it matters little in ordinary calculations which of these numbers is used, but considerable confusion exists through different numbers having been used in calculating quantities required for insertion in tables giving the properties of steam.

EXERCISES ON CHAPTER III.

1. Distinguish clearly between heat and temperature.

2. Express 42·4 B.T.U. in lb.-degree-Cent. units. Express the same quantity of heat in gram-calorie units.

3. Define "specific heat of a substance." A copper vessel weighs $4\frac{1}{2}$ lbs. Calculate the quantity of heat required to raise its temperature 80° F. Take the specific heat of copper to be 0·092.

4. Explain the different ways in which heat may be transferred Give examples.

5. 5 gallons of water at a temperature of 180° F. are poured into a tank containing 30 gallons of water at 60° F. Calculate the final temperature of the water on the assumption that no heat is lost.

6. A piece of copper weighing 2 lbs. is brought to a temperature of 212° F. and then dropped into a vessel containing 3 lbs. of water at 55° F. Neglecting any sources of loss, what will be the final temperature?

7. Answer Exercise 6 again on the supposition that the water equivalent of the vessel is 0·5 lb.

8. Give a brief explanation of our reasons for believing that heat is a form of energy.

9. Taking Joule's mechanical equivalent of heat to be 778 ft. lbs., calculate the mechanical work equivalent to the heat which must be imparted to a pound of water in order to raise its temperature from freezing point to boiling point.

10. A piece of iron weighing 100 grams is allowed to remain in a current of hot gases for a few minutes, and is then dropped into a calorimeter containing 500 c.c. of water at 15° C. The water equivalent of the calorimeter is 40 grams. The final steady temperature was observed to be 22°·5 C. Calculate the temperature of the gases. Take the specific heat of iron to be 0·1098.

11. One pound of coal when completely burned can give out 15,000 B.T.U.; one pound of petroleum, 20,500 B.T.U.; one cubic foot of lighting gas, 600 B.T.U. Express these quantities of heat in foot lbs. Take $J = 778$ ft. lbs.

12. The metal of a certain boiler weighs 13 tons and contains 11 tons of water. Calculate the quantity of heat in B.T.U. which must be supplied in order to raise the temperature of the metal and the water from 60° F. to 340° F. Take the specific heat of the metal to be 0·1098.

CHAPTER IV.

PROPERTIES OF GASES.

The gaseous state.—A substance in the gaseous state possesses the property of **indefinite expansion.** A small quantity of gas introduced into a perfectly empty vessel will at once expand and occupy the whole of the interior. Gases may exist either as **vapours**, or as so-called **perfect gases.** The perfect gas was supposed to exist under all conditions of pressure and temperature as a gas, but it is now well known that all gases can be liquefied by great pressure and cold. A vapour may be defined as a gas near its liquefying point, and a perfect gas as the same substance far removed from its liquefying point. Gases, such as oxygen, hydrogen, nitrogen, and atmospheric air (which is a mixture of oxygen and nitrogen) behave as perfect gases under ordinary atmospheric conditions of pressure and temperature. Steam as it comes from boiling water is a vapour, but, if heated to a high temperature after leaving the water, it behaves more like a perfect gas. If the temperature of any gas be raised, keeping the pressure constant, the volume will be increased; and, if the volume be kept constant, the pressure will be increased as the temperature is raised.

Pressure of the atmosphere.—The pressure of the atmosphere will be made evident and may be roughly measured by the following experiment:

EXPT. 11.—Take a glass tube closed at one end and open at the other, about 36″ long, and fill it with mercury. Close the open end by means of a finger, and invert the tube several times so as to collect into one bubble any air contained in the tube. Let this air escape

and add mercury so as to fill the tube to the top. Close the end again with a finger and invert the tube, placing the open end in a cup of mercury. On removing the finger, the tube being held vertically, the level of the mercury inside the tube will fall until it stands at a height h inches above that in the cup (Fig. 37).

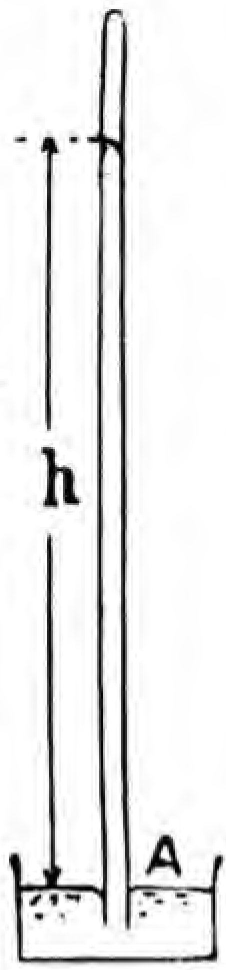


FIG. 37.—Apparatus for showing the principle of the barometer.

At A, the pressure inside the tube is that due to a column of mercury h inches high, and is equal to $(w \times h)$ lbs. per square inch, w being the weight of a cubic inch of mercury. The pressure of the atmosphere on the surface of the mercury in the cup will be equal to this. The average height of the mercury column is 30 inches, and as mercury weighs nearly 0·49 pound per cubic inch, this column represents a pressure of 0·49 multiplied by 30, or 14·7 pounds per square inch.

The apparatus constitutes the **common barometer.** It is useful to remember that every inch of mercury height in a barometer corresponds nearly to a pressure of $\frac{1}{2}$ lb. per square inch.

EXPT. 12.—Measure in your rough barometer, the height of the mercury column in inches and calculate from this the atmospheric pressure in lbs. per square inch. Compare also the height measured with that shown by a standard barometer at the same time.

Other forms of barometer.—A standard barometer is much more elaborately fitted than the rough one of Expt. 11. In Fig. 38, **Fortin's barometer** is illustrated. This instrument has a screw A fitted to the mercury cup by means of which the level of the mercury in the cup R may be brought to coincide with a point P. The upper part of the case is furnished with a scale and a sliding vernier, operated by means of a thumb screw B. Mirrors are fitted to the back board behind the cup R and the scale S. To read this instrument, first adjust the level of the mercury in the cup R, using the screw A. Bring the eye to the level of the top of the mercury column (the mirror aids this) and operate the

screw *B* until the top of the vernier coincides with the top of the mercury column. Readings of the scale and vernier are then taken.

In the **aneroid barometer** a metallic box has one side made flexible. The box is exhausted of air as nearly as possible, and the flexible side will now be more or less forced in, as the magnitude of the external atmospheric pressure increases or diminishes. This movement of the side is communicated and magnified by means of levers and toothed wheels to a small spindle which projects outside the case and carries an index finger which is thus caused to travel over a scale. By suitably graduating the scale the pressure of the atmosphere will be indicated by the position of the index finger.

FIG. 38.—Fortin's barometer.

Measurement of pressure of a gas.—Engineers in this country usually measure gaseous pressures in pounds per square inch; or, if the pressure be very high, in atmospheres, one atmosphere being a gaseous pressure of 14·7 lbs. per square inch. For low pressures of steam such as are found in condensers, the pressure is usually stated in inches height of mercury column, as this facilitates comparison with the barometric pressure. The metric unit of gaseous pressure is usually one kilogram per square centimetre. Taking 1 square inch = 6·45 square cms., and 1 kilogram = 2·205 lbs., a pressure of 1 kilogram per square centimetre will be equivalent to $(6{\cdot}45 \times 2{\cdot}205) = 14{\cdot}22$ lbs. per square inch. For rough conversion from the metric to the British system, it is convenient to remember that a pressure of one kilogram per square centimetre is approximately equal to a pressure of one atmosphere (*i.e.* 14·7 lbs. per square inch).

Chimney draught, which is the difference in the gaseous pressures inside and outside at the base

of a chimney, is usually measured in inches of water. A column of water 144 feet high and 1 square inch in section (*i.e.* one cubic foot in volume) gives a pressure at its base of 62·4 lbs. Hence a column 2·3 feet high gives 1 lb. per square inch pressure. 1 inch water pressure = 0·036 lb. per square inch = 5·2 lbs. per square foot.

There are two zeros of gaseous pressure from which other pressures may be measured; these are:

(*a*) atmospheric pressure, other pressures being stated as so much above or below this;

(*b*) perfect vacuum, that is the condition of pressure which exists in a space perfectly empty of gas, which, of course, will be devoid of all gaseous pressure.

Pressures stated above perfect vacuum are said to be **absolute pressures**; those stated from atmospheric pressure are said to be **gauge** or **bursting pressures**. The meaning of the last term will be rendered evident by considering a vessel containing gas under a pressure of say 100 lbs. per square inch above atmospheric pressure. The interior walls of the vessel will actually be subjected to a pressure of (100+14·7)=114·7 lbs. per square inch (absolute pressure), which pressure, tending to force the walls of the vessel outwards, is partially counteracted by the external pressure of the atmosphere, 14·7 lbs. per square inch, tending to crush or collapse the vessel. The net pressure tending to burst the vessel will be the difference, *i.e.* 100 lbs. per square inch, which is therefore called the bursting pressure. The name gauge pressure given to the same pressure is due to the fact that gauges used for indicating the gaseous pressure in a closed vessel show, not the absolute pressure, but the difference between the absolute pressure inside the vessel and the atmospheric pressure outside. Pressure gauges will be described later.

Boyle's Law for perfect gases.—The experiments of Boyle and others, on the connection between pressure and volume of gases, show that **the absolute pressure varies inversely as the volume, provided the temperature remains unaltered.** Taking a given mass of gas under conditions of pressure and volume p_1 and v_1, let these be changed, without alteration of temperature, to any other conditions p_2 and v_2, then

$$p_1 : p_2 = v_2 : v_1,$$

or,

$$p_1 v_1 = p_2 v_2.$$

Since any other conditions of pressure and volume may be taken, we may write the law as

$$pv = \text{a constant.}$$

This law has been proved experimentally to be followed closely by such gases as air, hydrogen, oxygen, nitrogen and others when not very far removed from ordinary atmospheric conditions of pressure and temperature, but it is not followed by steam and other vapours which are near to their liquefying points.

A perfect gas is sometimes defined as one which closely obeys Boyle's Law.

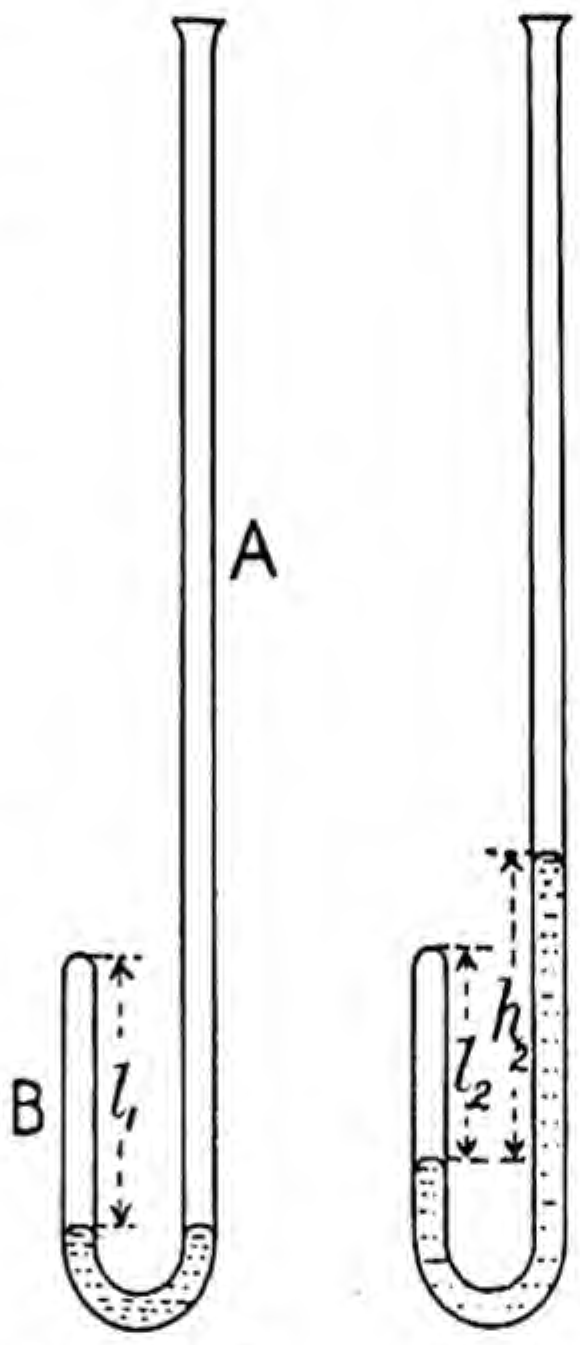


FIG. 39.—Tube used in verifying Boyle's Law.

FIG. 40.

EXPT. 13.—To verify Boyle's Law roughly, the apparatus shown in Fig. 39 may be used. A vertical glass tube is bent as shown, the long limb A being left open and the short limb B closed. Mercury is introduced and adjusted so that it stands at the same level in both limbs. The air enclosed at B will then be at the atmospheric pressure shown by a barometer. Read the barometer and let its height be h_1 inches of mercury. The volume of the enclosed air may be stated with sufficient accuracy by measuring the length of tube occupied by air and taking this length to represent the volume. Let this be l_1. On pouring more mercury into A, the level in the long limb will be found to be higher than that in B (Fig. 40). Let h_2 be the difference in levels. Then the pressure of the enclosed air will be $(h_1 + h_2)$, and its volume will be represented by l_2. Repeat the experiment several times, in each case waiting a minute or so after adding more mercury in

order to allow the compressed air to cool to the temperature of the room, and tabulate thus:

Pressure, inches of mercury.	Volume, length of tube *B* occupied by air.	Product of pressure and volume.

It will be found that the products in the last column are very nearly equal to one another and to the first product $h_1 l_1$. Plot columns 1 and 2, and draw a curve to illustrate Boyle's law (Fig. 41).

Mathematicians call this curve a **rectangular hyperbola.** As it is of considerable importance, two methods of finding points in the curve from given particulars will now be explained.

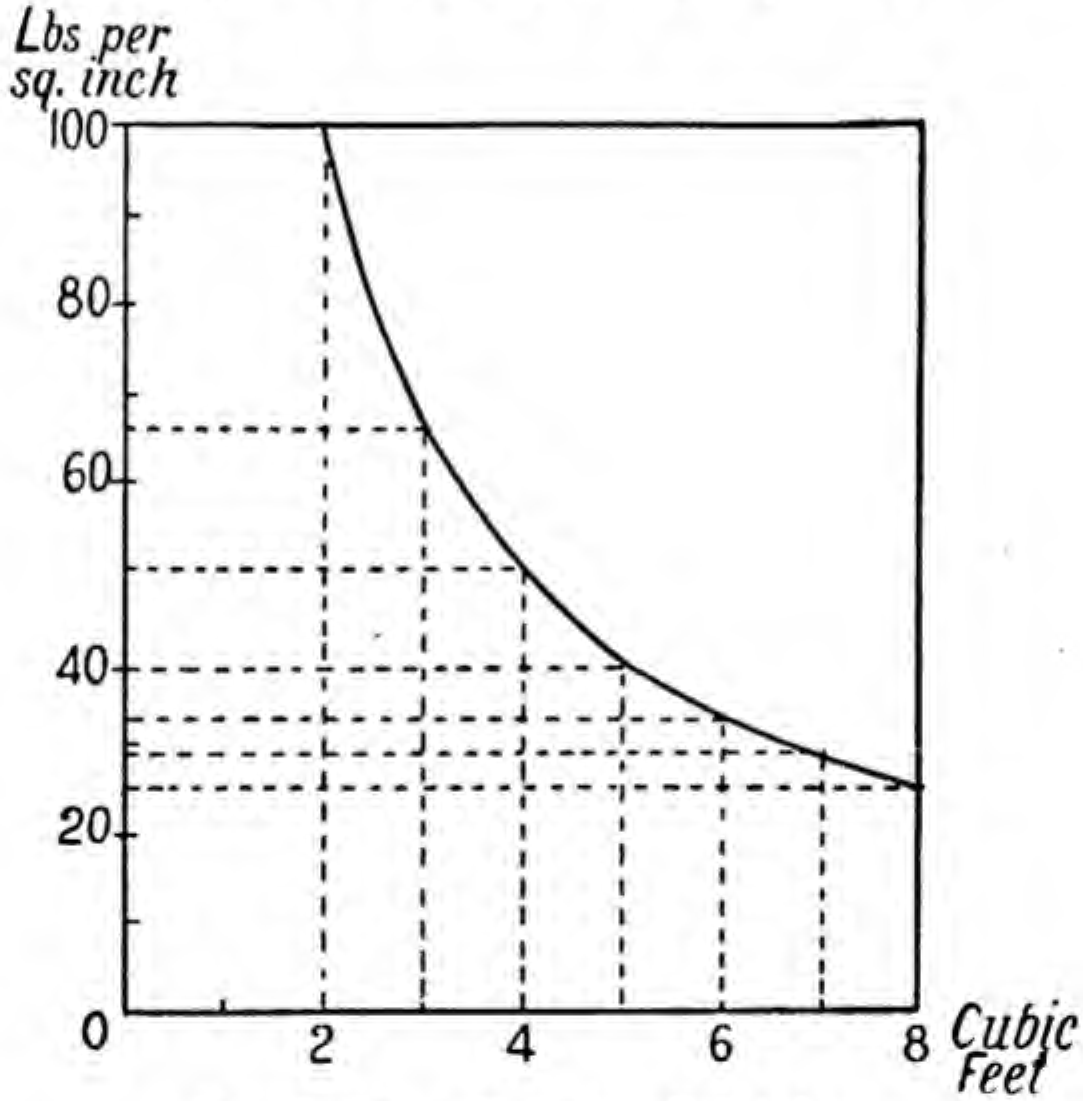


FIG. 41.—Curve illustrating Boyle's Law.

EXAMPLE. 2 cubic feet of air at an absolute pressure of 100 lbs. per square inch are expanded at uniform temperature until the volume occupied is 8 cubic feet. Plot a curve showing the expansion.

Method 1. From Boyle's law,

$$p_1v_1 = p_2v_2 = p_3v_3 = \text{etc.} \qquad (1)$$

Take volumes differing by 1 cubic foot, and **calculate** corresponding pressures, using equations (1) which will take the form

$$p_2 = \frac{p_1v_1}{v_2};\ p_3 = \frac{p_1v_1}{v_3},\ \text{etc.}$$

Arrange the results in a table, thus:

Volumes, cubic feet.	Pressures, lbs. per sq. inch.	Volumes, cubic feet.	Pressures, lbs. per sq. inch.
2	100	6	33·3
3	66·6	7	28·6
4	50	8	25
5	40		

Plot these as shown in Fig. 41.

Method 2. Take two axes OX, OY (Fig. 42). Set off OA along OY to represent p_1, and OC along OX to represent v_1 to convenient scales of pressure and volume. Make $OE = v_2$. Complete the rectangles $OABC$ and $OADE$. Join OD cutting CB in F. Draw FG parallel to OX. Then EG is equal to p_2, so that G will be a point on the expansion curve.

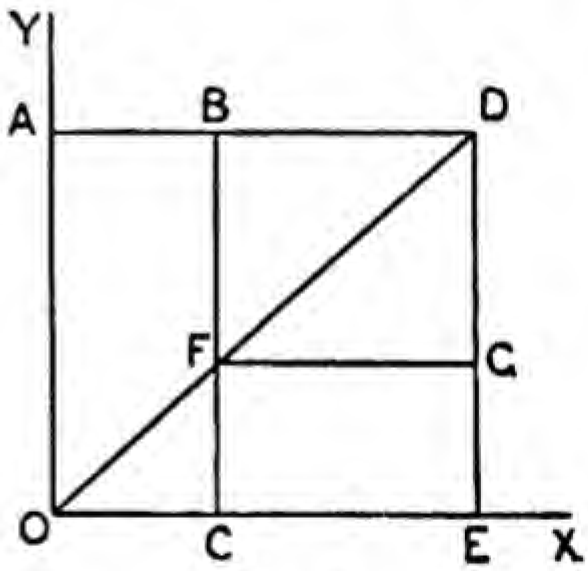


Fig. 42.

Proof. The triangles OCF and OED are similar, therefore

$$FC : OC = DE : OE,$$

or, $$EG : OC = BC : OE.$$

But $$OC = v_1,\ BC = p_1,\ \text{and}\ OE = v_2.$$

$$\therefore\ EG : v_1 = p_1 : v_2,$$

or, $$EG = \frac{p_1v_1}{v_2},$$

and therefore, by Boyle's law, $EG = p_2$.

Other points may be found in a similar fashion. Fig. 43 shows the complete curve obtained by the use of this constructional method.

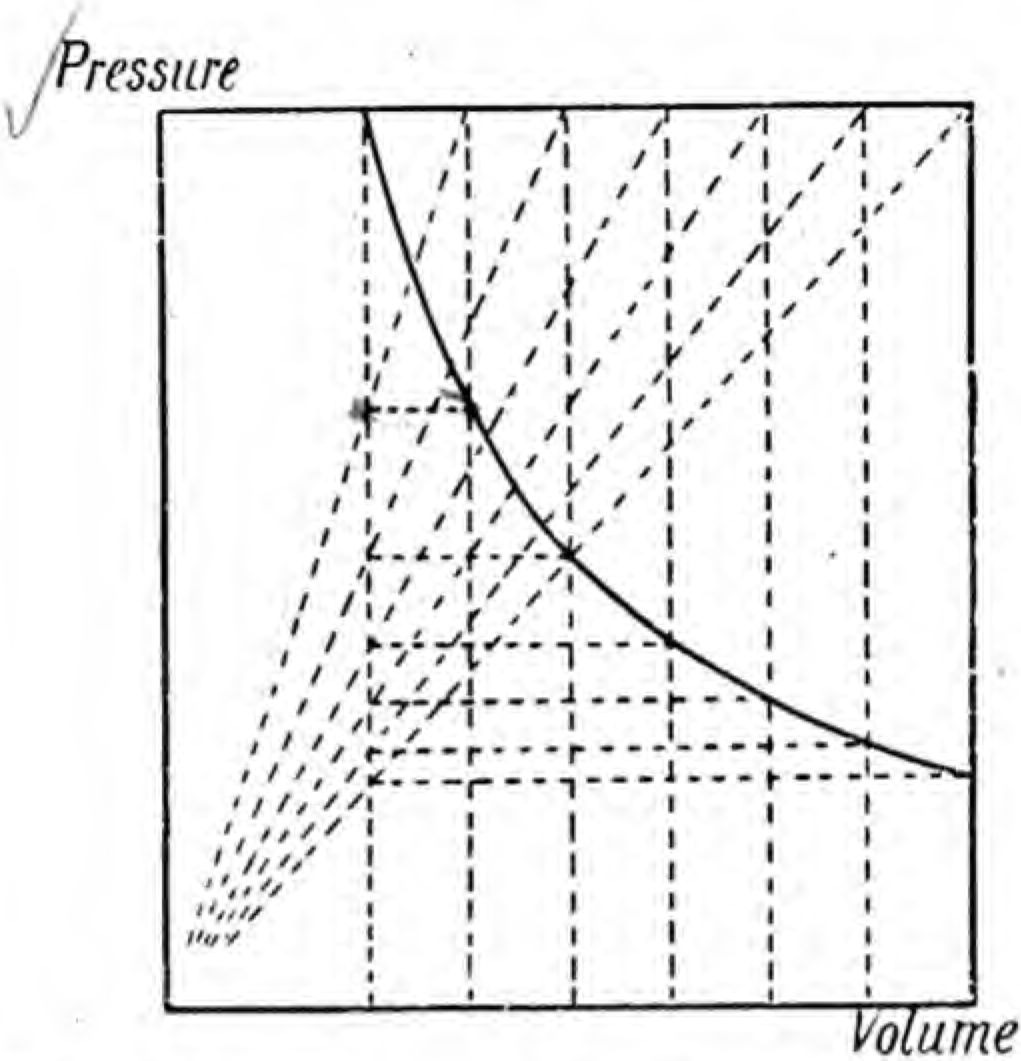


FIG. 43.—Boyle's Law curve drawn by means of a geometrical construction.

Charles's law.—This law, first enunciated by Charles and Gay Lussac, states that **all perfect gases expand by the same fraction of the volume they occupy at freezing temperature when their temperature is raised one degree, provided the pressure remains unaltered.**

Using the Centigrade scale of temperature, the fraction has been found experimentally to be $\frac{1}{273 \cdot 7}$, or $\left(\frac{1}{273 \cdot 7} \times \frac{5}{9}\right) = \frac{1}{493}$ if the Fahrenheit scale be employed.

Let V_0 = volume of a given mass of gas at freezing temperature;

V = its volume at any other temperature t, the pressure being unaltered.

Then,

$$V = V_0 + \frac{1}{273 \cdot 7} V_0 t, \text{ where } t \text{ is the Centigrade temperature,}$$

or,

$$V = V_0 + \frac{1}{493} V_0 (t - 32), \text{ where } t \text{ is the Fahrenheit temperature.}$$

These equations are not convenient for calculation, as usually the volume stated is given at some other temperature than freezing. To use the above equations, the volume occupied at freezing temperature would have to be calculated first, before the volume V occupied at some stated temperature t could be found.

Absolute scale of temperature.—By use of the absolute scale of temperature, calculations on the changes of volume and temperature of a gas become much simpler. Suppose we take 493 cubic feet of a gas at freezing temperature, 32° F., and, at constant pressure, raise the temperature to 33° F. The volume will become

$$\{493+(\tfrac{1}{493}\times 493)\}=494 \text{ cubic feet.}$$

Raising the temperature to 34° F. will give a volume of

$$\{493+(\tfrac{2}{493}\times 493)\}=495 \text{ cubic feet.}$$

At 35° F. the volume would become 496 cubic feet and so on, each degree Fah. of increase in the temperature producing an increase of 1 cubic foot in the volume. Regraduate the Fahrenheit thermometer by marking freezing point 493° instead of 32° and boiling point 673° instead of 212°, and it is easy to see that now the volumes occupied by the gas will be proportional to the temperatures on the regraduated scale. Such a scale is called an **absolute scale of temperature.** Had a Centigrade thermometer been employed, we should have placed 273·7° at freezing point and 373·7° at boiling point. These scales are referred to as the **absolute temperature Fah.**, and the **absolute temperature Cent.** respectively.

We shall use the letter t for ordinary temperatures, Fahrenheit or Centigrade, and the Greek letter τ for absolute temperatures. Zero on the ordinary Fahrenheit scale is marked $(493-32)=461°$ on the absolute scale.

To convert from the ordinary scales to the absolute scales we have:

$$\tau=461+t° \text{ F. for the Fah. scale.}$$

$$\tau=273{\cdot}7+t° \text{ C. for the Cent. scale.}$$

Charles's law may now be stated thus:

The volumes of all perfect gases are proportional to the absolute temperatures at which they are measured, provided the pressure remains unaltered.

Let V_1 = volume of a given mass of gas at absolute temperature τ_1.

V_2 = volume of the same mass at absolute temperature τ_2, and at the same pressure.

Then, $$V_1 : V_2 = \tau_1 : \tau_2,$$

or, $$V_1\tau_2 = V_2\tau_1,$$

or, $$\frac{V_1}{\tau_1} = \frac{V_2}{\tau_2}.$$

This equation is suitable for use in calculations.

Example i. One pound weight of air at 0° C. and 14·7 lbs. per square inch pressure occupies a volume of 12·4 cubic feet. Find the volume of the same weight of air when the temperature is 60° F., the pressure being unaltered.

This question may be worked using the absolute Fah. scale.

$$V_1 = 12{\cdot}4 \text{ cubic feet.}$$
$$\tau_1 = 32 + 461 = 493° \text{ F. abs.}$$
$$\tau_2 = 60 + 461 = 521° \text{ F. abs.}$$
$$\frac{V_1}{\tau_1} = \frac{V_2}{\tau_2}.$$
$$\therefore\ V_2 = \frac{V_1\tau_2}{\tau_1}$$
$$= \frac{12{\cdot}4 \times 521}{493}$$
$$= \underline{13{\cdot}1} \text{ cubic feet.}$$

It is convenient to remember that the volume of one pound weight of air under ordinary conditions of atmospheric pressure and temperature is 13 cubic feet nearly.

Example ii. In a certain boiler test it was found that 300 cubic feet of air at a pressure of one atmosphere and 60° F. temperature entered the furnace per lb. of coal burned. What will be the weight of air admitted per lb. of coal?

Using the approximate number found in Example i.:

$$\left.\begin{array}{r}\text{Weight of air}\\ \text{per lb. of coal}\end{array}\right\} = \frac{300}{13} = \underline{23 \text{ lbs.}} \text{ nearly.}$$

Example iii. A room measures 50 feet × 30 feet × 25 feet. If the air in it be heated from 40° F. to 60° F. what percentage of the contained air will be expelled?

$$V_1 = 50 \times 30 \times 25 = 37{,}500 \text{ cubic feet.}$$
$$\tau_1 = 461 + 40 \quad = 501° \text{ F. abs.}$$
$$\tau_2 = 461 + 60 \quad = 521° \text{ F. abs.}$$

Volume of the original air in the room if heated to 60° F. $= V_2 = \dfrac{V_1\tau_2}{\tau_1}$

$$= \frac{37{,}500 \times 521}{501}$$

$$= \underline{38{,}997} \text{ cubic feet.}$$

Volume of air expelled $= 38{,}997 - 37{,}500$

$$= \underline{1497} \text{ cubic feet.}$$

Notice that the volume of air expelled is stated at a temperature of 60° F.

$$\text{Percentage expelled} = \frac{1497}{38{,}997} \times 100$$

$$= \underline{3{\cdot}8}.$$

Combination of Boyle's and Charles's laws.—It has now been seen that p is inversely proportional to v when t is constant, and that v is directly proportional to τ when p is constant. A law must now be found which applies when all three conditions, p, v, τ, vary simultaneously. Such a law might be written down at once from the algebraic rules of variation, but we proceed to find it in the following manner.

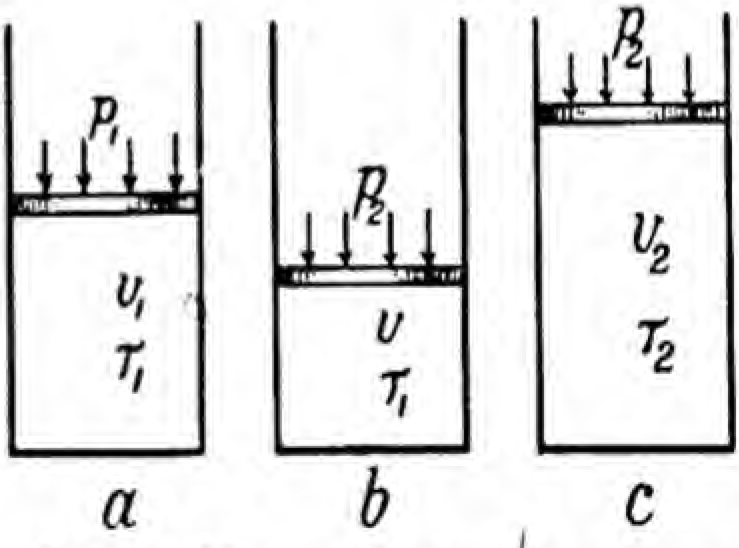


FIG. 44.—Diagram showing changes occurring in the pressure, volume, and temperature of a gas.

Let a mass of gas be inclosed in a cylinder, Fig. 44 (a), under given conditions p_1, v_1, τ_1. Suppose first that p_1 and v_1 are changed at constant temperature τ_1 until a pressure of p_2 and a volume v are obtained.

Applying Boyle's law
$$p_1v_1 = p_2v,$$

$$v = \frac{p_1v_1}{p_2} \quad \text{.................................} \quad (1)$$

The gas will now be as shown at (b) in Fig. 44.

Now change the temperature from τ_1 to τ_2 keeping the pressure constant at p_2. The volume will change from v to v_2 as shown at (c) in Fig. 44.

Applying Charles's law
$$\frac{v}{\tau_1}=\frac{v_2}{\tau_2};$$
$$\therefore\ v=\frac{v_2\tau_1}{\tau_2}\quad\text{.................................. (2)}$$

The results obtained in (1) and (2) being now equated, we obtain
$$\frac{p_1v_1}{p_2}=\frac{v_2\tau_1}{\tau_2},$$

or,
$$\frac{p_1v_1}{\tau_1}=\frac{p_2v_2}{\tau_2}\quad\text{.................................. (3)}$$

Evidently had the conditions been varied simultaneously instead of in the step by step manner adopted above, we should have obtained the same result.

Writing it in the proportional form :
$$p_1v_1 : p_2v_2=\tau_1 : \tau_2,\quad\text{.............................. (4)}$$
which may be stated thus : When any operation is performed on a given mass of gas involving changes in the pressure, volume and temperature, **the product of the absolute pressure and the volume is proportional to the absolute temperature.**

The law so found may also be written
$$pv=\text{a constant }\times\tau.$$
Writing c for the constant,
$$pv=c\tau\quad\text{.. (5)}$$

This equation is sometimes called the characteristic equation of a gas. It will be remembered that in this form it must not be applied to vapours which do not obey Boyle's and Charles's laws.

Equation for air.—In finding the characteristic equation, it is customary to measure the pressure in lbs. per square foot, the volume is taken as that occupied by one lb. weight of the gas at some temperature τ. For air, we may take

$P=14{\cdot}7\times144=2{,}116$ lbs. per square foot.

$V=12{\cdot}4$ cubic feet at 32° F. and pressure 14·7 lbs. per square inch.

$\tau=461+32=493°$ F. abs.
$$PV=c\tau.$$
$$\therefore\ c=\frac{PV}{\tau}$$
$$=\frac{2116\times12{\cdot}4}{493}$$
$$=\underline{53{\cdot}2}.$$

The characteristic equation for air is, therefore,

$$PV = 53{\cdot}2\tau.$$

Isothermal and adiabatic expansion.—A gas expanding or being compressed at constant temperature, is said to do so **isothermally**, that is, at equal or constant temperature. Any operation conducted at constant temperature is said to be an **isothermal operation.** A curve such as is plotted in Fig. 41 in illustration of Boyle's law (where the temperature is kept constant) is called an **isothermal curve.**

A gas expanding or being compressed in such a manner that no heat is allowed to enter it or escape from it while the operation is being conducted is said to be undergoing **adiabatic expansion** or **adiabatic compression.** Such an operation may be imagined if conducted in a cylinder, the walls and piston of which have no capacity for heat, *i.e.* are unable to absorb any heat, and are also perfect non-conductors. Needless to say, such a cylinder cannot be constructed, and consequently such expansion cannot be realised in a cylinder. A near approach to adiabatic expansion is obtained when steam is allowed to flow freely through a nozzle, expanding as it does so (p. 319).

An isothermal operation may be imagined in the following manner. Suppose we have a mass of air inclosed in a cylinder the walls of which are good conductors. Maintain the temperature of the room constant throughout, and allow the piston to move out very slowly, taking, say a day or a week, to perform a stroke of one foot length. There is thus plenty of time for heat to flow through the cylinder walls and so maintain the temperature of the inclosed air constantly at that of the air in the room.

Relation of pressure and temperature in a gas.—Taking equation (3) above (p. 54), $\frac{p_1 v_1}{\tau_1} = \frac{p_2 v_2}{\tau_2}$

and put $v_1 = v_2$, thus causing the pressure and temperature to change at constant volume. This will give the law

$$\frac{p_1}{\tau_1} = \frac{p_2}{\tau_2},$$

or,

$$p_1 : p_2 = \tau_1 : \tau_2.$$

That is, **the pressure of a perfect gas is directly proportional to the absolute temperature, the volume being kept constant.**

Specific heats of a gas.—In Fig. 45 is shown a cylinder fitted with a piston which may be loaded to produce any constant pressure on the gas. Application of heat to the gas will produce two effects,

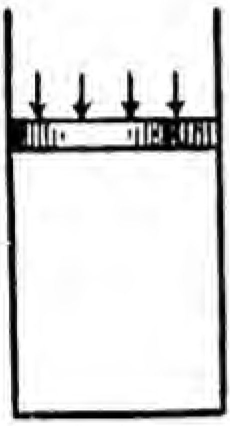


Fig. 45.

i. The gas will have its temperature raised.

ii. It will expand, driving the piston upwards, and thus do work against the resistance of the applied pressure.

Now allow the gas to return to its original conditions; then fix the piston, and apply heat so as to raise the temperature of the gas to the same extent as before. This latter operation being conducted at constant volume, and the piston not moving, no work is done against the applied pressure. In the first case, sufficient heat must be given to raise the temperature of the gas, and in addition an amount of heat equivalent to the external mechanical work done against the resistance. In the second case, only an amount of heat sufficient to raise the temperature of the gas need be imparted.

It follows therefore that **the specific heat of a gas at constant pressure is greater than the specific heat of the same gas at constant volume.** The specific heat of air at constant pressure is 0·238, and at constant volume is 0·169. Thus, to raise the temperature of 1 lb. weight of air through 1° F. requires 0·238 B.T.U. at constant pressure, and 0·169 B.T.U. at constant volume.

EXERCISES ON CHAPTER IV.

1. Distinguish between a perfect gas and a vapour; also between the gauge pressure and absolute pressure of a gas.

2. State Boyle's law. Under what conditions is Boyle's law followed by a gas?

3. A machine for compressing air takes in 4 cubic feet from the atmosphere at a pressure of 14·7 lbs. per square inch absolute, and compresses it to a pressure of 100 lbs. per square inch as shown by the gauge. Assuming the temperature to remain constant, calculate the final volume. Draw a curve showing the changes of pressure and volume.

4. A bicycle tyre has a capacity of 150 cubic inches when fully inflated, the pressure being then 25 lbs. per square inch by gauge.

Supposing the tyre to be quite flat at first, calculate what volume of atmospheric air must be used in order to inflate it.

5. State Charles's law. Explain the reason for using an absolute scale of temperature in applying this law.

6. A boiler chimney is 80 feet high and has internal dimensions 2 feet × 2 feet square. Suppose the temperature of the gases inside to be 500° F.; calculate what volume the contents of the chimney would occupy if the temperature were reduced to 60° F. without change in pressure.

7. A cylinder fitted with a piston contains at a certain instant 6 cubic feet of gas at 15 lbs. per square inch absolute and 15° C. Operations are being conducted on the gas involving changes in pressure, volume and temperature, the weight of gas present being kept constant. At another instant, the pressure is found to be 150 lbs. per square inch absolute and the volume 2·5 cubic feet. Calculate the temperature at this instant.

CHAPTER V.

PROPERTIES OF STEAM.

Water.—Water is a chemical compound of hydrogen and oxygen. This fact may be shown by mixing together two volumes of hydrogen gas with one volume of oxygen gas and exploding them. The gases do not unite until the mixture is ignited, when an explosion occurs; the hydrogen and oxygen unite, forming a compound which cannot be separated into its constituents by simple mechanical means. This compound is water vapour, *i.e.* **steam**, which on cooling condenses into ordinary water. As the experiment is dangerous, the student is recommended not to carry it out himself.

States of water.—Water may exist in the solid state as ice, in the liquid state as ordinary water, as a vapour—ordinary steam, and as a perfect gas if the vapour be raised to a moderately high temperature at constant pressure. Further elevation of the temperature causes the hydrogen and oxygen to dissociate, that is, the molecules of hydrogen and oxygen part company, remaining as a mixture of these two gases until the temperature falls somewhat, when they reunite to form steam.

To cause water to pass successively through the states in the order named, requires heat to be imparted; and, similarly, heat must be abstracted if changes in the reverse order are to be effected.

Sensible heat.—It has already been seen that to raise the temperature of one pound of water through 1° F. requires 1 B.T.U. Heat imparted to a substance which produces a rise of temperature is called **sensible heat**. The fact that sensible heat is entering a substance will, of course, be always rendered evident by a thermo-

meter. It is customary to reckon the sensible heat of water from 32° F. To raise the temperature of one pound of water from 32° F. to 212° F. requires 180·5 B.T.U., which number therefore gives the sensible heat of water at 212° F. Notice that the sensible heat of water at 212° F. would be (212 − 32) = 180 B.T.U. if the specific heat of water were unity at all temperatures, but owing to slight variations in the specific heat the number stated is more accurate. We may assume for ordinary calculations that the specific heat of water does remain constant, in which case the sensible heat, h, required to raise the temperature of w lbs. of water from a temperature t_1° F. to a temperature t_2° F. will be found from

$$h = w(t_2 - t_1) \text{ in B.T.U.}$$

If greater accuracy be required, the table giving values of h (p. 454) may be consulted.

Latent Heat.—In testing the fixed points of a thermometer, the student has noticed that the thermometer remains steady throughout the melting of ice and the boiling of water, in spite of the fact that heat is being continually imparted to these substances. The conclusion arrived at is that heat must be imparted to cause water to change its state from solid to liquid or from liquid to gaseous. **Heat imparted to a substance and producing a change of state without change of temperature is called Latent Heat.** If the change of state is from gaseous to liquid, or from liquid to solid, latent heat must be abstracted from the substance.

EXPT. 14.—Weigh a copper calorimeter and pour in about ½ gallon of water at a temperature of about 100° F. Weigh again and find the exact weight W of the water by subtraction. Take a piece of ice weighing about ½ lb. Wipe any water from its surface, take the temperature t_1° F. of the water in the calorimeter and plunge in the ice. Keep stirring until the ice is melted and note the temperature t_2° F. at the instant the last piece of ice disappears. Weigh again and subtract from the total weight that of the calorimeter and of the water originally in it, obtaining w, the weight of ice which has been melted.

Assuming that the heat abstracted from the original water has been used altogether

(a) in changing the state of the ice from solid to liquid;

(b) in elevating the temperature of the resulting water from freezing point to t_2° F.;

we have, calling the latent heat of 1 lb. of ice l, in B.T.U.

$$W(t_1 - t_2) = lw + w(t_2 - 32^\circ);$$

$$\therefore\ l = \frac{W(t_1 - t_2) - w(t_2 - 32^\circ)}{w}.$$

Careful experiments show that the latent heat of 1 lb. of ice is 144 B.T.U. Compare this with your experimental result. Do they agree fairly closely if the water equivalent of the calorimeter be taken into account? If not, how do you explain the discrepancy?

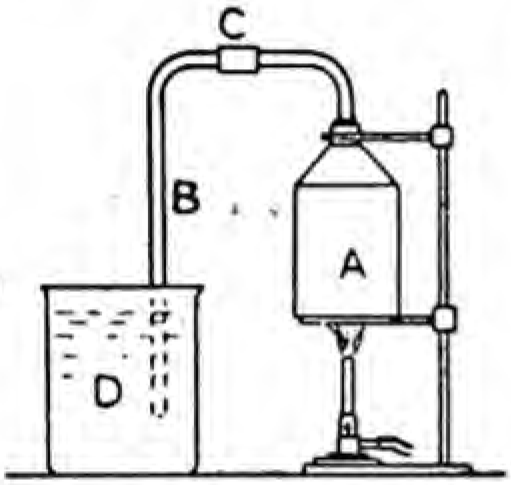


FIG. 46.—Apparatus for determining the latent heat of steam at atmospheric pressure.

EXPT. 15.—Arrange the apparatus as shown in Fig. 46. A is a copper vessel about 5″ diameter × 9″ high, serving as a boiler. Steam is taken from it by means of a glass tube B, about ¼″ bore, connected to the boiler by means of a cork. B is best made in two pieces connected at C by a piece of rubber tube about 1½″ long. The steam is discharged into a calorimeter D containing a known weight of water, and is condensed there, its latent heat and part of its sensible heat being given up to the water.

To measure the latent heat of steam at a pressure equal to that of the atmosphere by means of this apparatus requires the following quantities to be known:

W = weight of water originally in the calorimeter.
w = weight of steam condensed.
t_1 = original temperature of the water W.
t_2 = final temperature of the mixture consisting of W and w.
The water equivalent of the calorimeter.

First make a preliminary experiment. Bring the water in A to boiling temperature; and after a minute or two, when steam is being given off freely, take the temperature t_1 of the water in D, and then immerse the open end of the tube B in the water in D, noting the time as you do so. A crackling noise will be heard proceeding from D, due to bubbles of steam issuing from B and instantly collapsing on coming into contact with the cold water. While the experiment is going on for 10 minutes or so you may observe the following points. Closely examine the glass tube, when you will observe a film of water covering its inner surface and travelling towards D to be discharged

finally into the calorimeter. Notice also that if the boiler be strongly heated so that the ebullition is violent, a considerable quantity of water may enter the tube from the boiler. This effect will be magnified if there is too much water in the boiler, thus restricting the volume in the boiler available for steam space.

Now it is evident that any water of condensation entering the calorimeter has already given up its latent heat to some body other than the water in the calorimeter, but as it will still be hot water at a temperature not much below 212° F., it will give up sensible heat to the water in the calorimeter. It would be an advantage if only **dry steam**, *i.e.* steam containing no suspended water entered the calorimeter, as then no correction for water carried over need be applied. We may approximate to this by using the following precautions:

(*a*) Violent ebullition and too restricted steam space cause **priming**, that is, a large quantity of water passes into the steam pipe from the boiler. Remedy—let the water boil quietly, reducing the flame if necessary, and do not have too much water in the boiler.

(*b*) Condensation takes place in the connecting pipe, which may be reduced by covering the pipe with strips of flannel.

(*c*) Any water now finding its way along the pipe may be separated from the steam by means of a separator. A simple form of separator may be arranged as in Fig. 47, where E is a small bottle fitted with a rubber stopper having two holes for receiving glass tubes. B is the tube coming from the boiler, and reaches about half-way down the bottle. B' is a tube connecting the separator to the calorimeter D; this tube terminates immediately on passing through the rubber stopper. Water coming along B will be deposited in the separator and very nearly dry steam will now be discharged into the calorimeter.

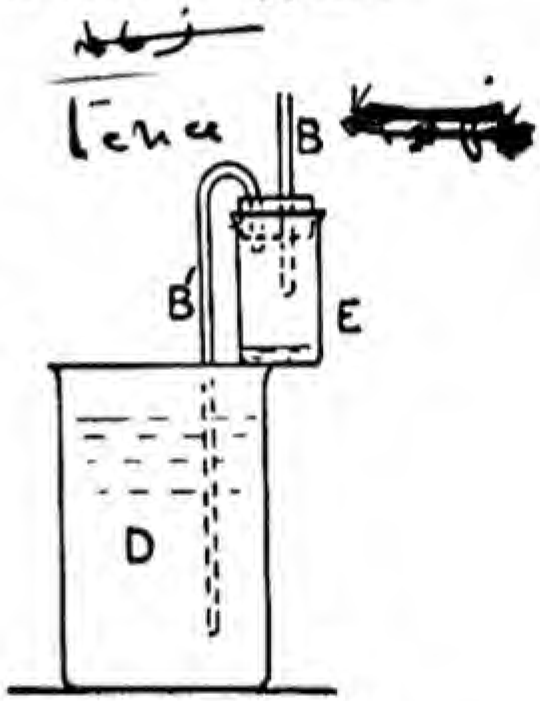


FIG. 47.—Arrangement for separating water from the steam.

Returning to the preliminary experiment, when the temperature of the water in the calorimeter has been raised 20° or 30° F., remove the supply tube, again noting the time, observe the temperature t_2 of the mixture in the calorimeter, and weigh the whole; deduct from this weight that of the calorimeter and of the water originally in it, the result being the weight of stuff w which has come from the boiler as a mixture of water and steam.

Calling the latent heat of 1 lb. of steam L, and neglecting meanwhile the water carried over, we may calculate L on the assumption that, as the weight w has parted with latent heat and with sensible heat in cooling from 212° F. to the final temperature t_2, the sum of these quantities of heat will be equal to that acquired by the water originally in the calorimeter. Hence the equations:

$$wL + w(212° - t_2) = W(t_2 - t_1) \qquad (1)$$

$$\therefore \ L = \frac{W(t_2 - t_1) - w(212 - t_2)}{w} \qquad (2)$$

The true result is about 967 B.T.U. Equation (2) will be found to give a result very different from this. Apply corrections to equation (1) as follows: (*a*) add to W, on the right hand side, the water equivalent of the calorimeter. Calling this e, the right hand side now becomes

$$(W + e)(t_2 - t_1);$$

(*b*) To correct for water carried over, allow the boiler to discharge steam from the open end of B into the atmosphere, raising the outer end of B for this purpose (Fig. 48). A beaker F will receive the drops of water carried over, the steam being blown off clear of the beaker. Allow this to go on during the same interval of time as the first experiment lasted, and it may be assumed that the weight of water collected is the same as that which has been carried into the calorimeter as water of condensation during the first experiment. Let this weight be called w_c, then of the whole weight w of stuff coming from the boiler, w_c is water and $(w - w_c)$ is steam. The left-hand side of equation (1) now becomes

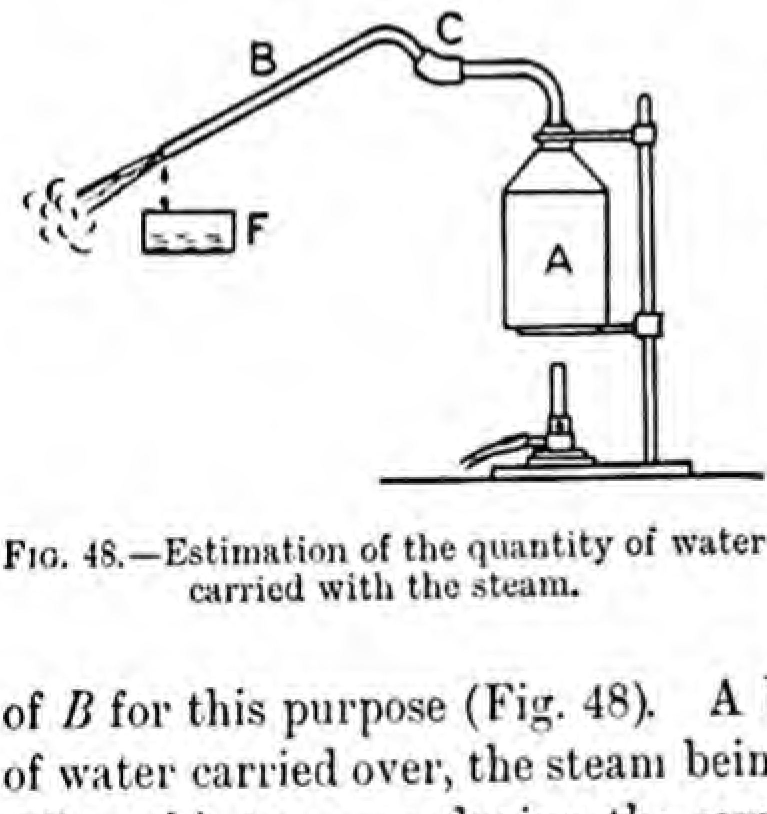


FIG. 48.—Estimation of the quantity of water carried with the steam.

$$(w - w_c)L + w(212° - t_2),$$

and the corrected equation will now be

$$(w - w_c)L + w(212° - t_2) = (W + e)(t_2 - t_1) \qquad (3)$$

$$\therefore \ L = \frac{(W + e)(t_2 - t_1) - w(212° - t_2)}{w - w_c} \qquad (4)$$

Compare the result for L as found from equation (4) with the true result 967 B.T.U. There should now be no serious difference.

Expt. 16.—Taking the precautions noted above regarding priming, covering the pipe and fitting a separator, repeat the experiment, making corrections for the water equivalent of the calorimeter and for any water carried past the separator, and compare the result with that found in the foregoing experiment.

Saturated and superheated steam.—When water is heated, boiling does not occur until a certain temperature is attained, depending on the pressure to which the water is subjected. Steam in contact with the water from which it has been formed is said to be **saturated steam.** Saturated steam at a given pressure can

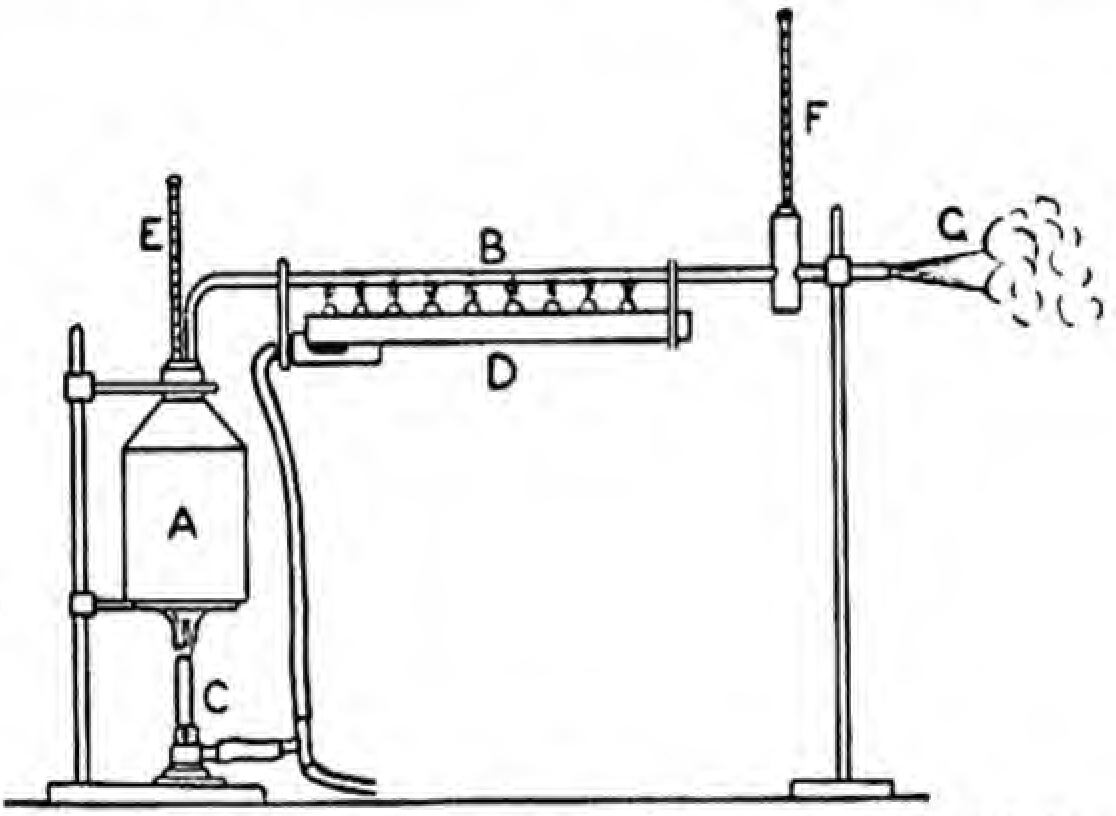


Fig. 49.—Apparatus for showing the temperature of superheated steam.

only exist at one temperature. Abstraction of heat from saturated steam does not produce a lowering of the temperature, but causes some of the steam to condense into water. Addition of heat to saturated steam, provided the pressure is maintained constant and no water is present, produces a rise in the temperature. The steam is no longer saturated steam, but is said to be **superheated.**

Superheated steam at a given pressure can exist at any temperature higher than the boiling temperature corresponding to that pressure. Saturated steam is a vapour, the pressure and temperature of which follow no simple law. Superheated steam behaves more and more like a perfect gas as its temperature is raised.

Expt. 17.—Fig. 49 shows a small copper boiler A, fitted with a cork having two holes, one of which serves for the insertion of a thermometer E, and from the other is led a copper tube B, having a pocket

formed near its outer end for the insertion of a thermometer F. D is a row of gas jets by means of which the tube B may be strongly heated. Put some water into the boiler and heat it. When steam has been given off freely for a few minutes read both thermometers. The pressure being practically atmospheric, both thermometers will read 212° F. Now light the row of gas jets and take readings of the thermometers. The pressure is still atmospheric, and the thermometer in the boiler will be found to read 212° F., but that in the pocket will be found to ascend to a temperature much higher than 212° F. The steam in the boiler is saturated steam at atmospheric pressure, that being discharged from the outer end of the tube is superheated steam at atmospheric pressure.

Pressure and temperature of saturated steam.—Experimental data due to Regnault show that the pressure and temperature of saturated steam are not simply proportional to one another. A Table showing corresponding values of these will be found on p. 454. From this Table it may be seen by plotting corresponding values of p and t (Fig. 51), that the pressure rises rapidly as the temperature increases. The relation cannot be deduced from first principles, but may be approximately presented by an empirical formula, *i.e.* one contrived so as to agree closely with the experimental data. Rankine's formula is:

$$\log_{10}p = 6{\cdot}1007 - \frac{B}{\tau} - \frac{C}{\tau^2},$$

where p = pressure in lbs. per square inch,

τ = absolute temperature Centigrade,

$$\log_{10}B = 3{\cdot}1812,$$
$$\log_{10}C = 5{\cdot}0881.$$

This formula may be used, in the absence of a Table, to calculate corresponding values of p and t.

EXAMPLE. What will be the pressure of saturated steam at a temperature of 200° C.? $\tau = (200 + 273{\cdot}7)$; $\log_{10}\tau = 2{\cdot}6755$.

$$\log_{10}\left(\frac{B}{\tau}\right) = 3{\cdot}1812 - 2{\cdot}6755$$
$$= 0{\cdot}5057;$$
$$\therefore \frac{B}{\tau} = 3{\cdot}204.$$
$$\log\left(\frac{C}{\tau^2}\right) = 5{\cdot}0881 - 2 \times 2{\cdot}6755$$
$$= \bar{1}{\cdot}7371;$$

$$\therefore \frac{C}{\tau^2}=0{\cdot}5459.$$

$$\log_{10}p=6{\cdot}1007-3{\cdot}204-0{\cdot}5459$$
$$=2{\cdot}3508;$$

$$\therefore p=\underline{224}{\cdot}3 \text{ lbs. per square inch.}$$

The tabular number will be found to be 225·9 lbs. per square inch, showing an error in the calculated result of about 0·7 per cent. It will be found better to use the Tables where possible. Any pressure intermediate to those given in the Table, p. 455, may be found readily by plotting the temperatures and pressures nearest to the given temperature.

EXPT. 18. To obtain the relation of pressure and temperature of saturated steam, the apparatus illustrated in Fig. 50 will be found useful. To use it a supply of steam from a steam boiler must be available. *A* is a strongly made vessel constructed of solid drawn brass tube about 4″ diameter, having brass flanges brazed on and brass covers bolted to the flanges. *B* is a coil made of ⅜″ copper tube. Steam from the boiler enters the coil through a regulating valve *C*, and is discharged through another valve *D*. Distilled water is poured into *A* through a valve *H* in quantity sufficient to cover the coil. *E* is a thermometer pocket, containing a thermometer *F*. *G* is a steam pressure gauge.

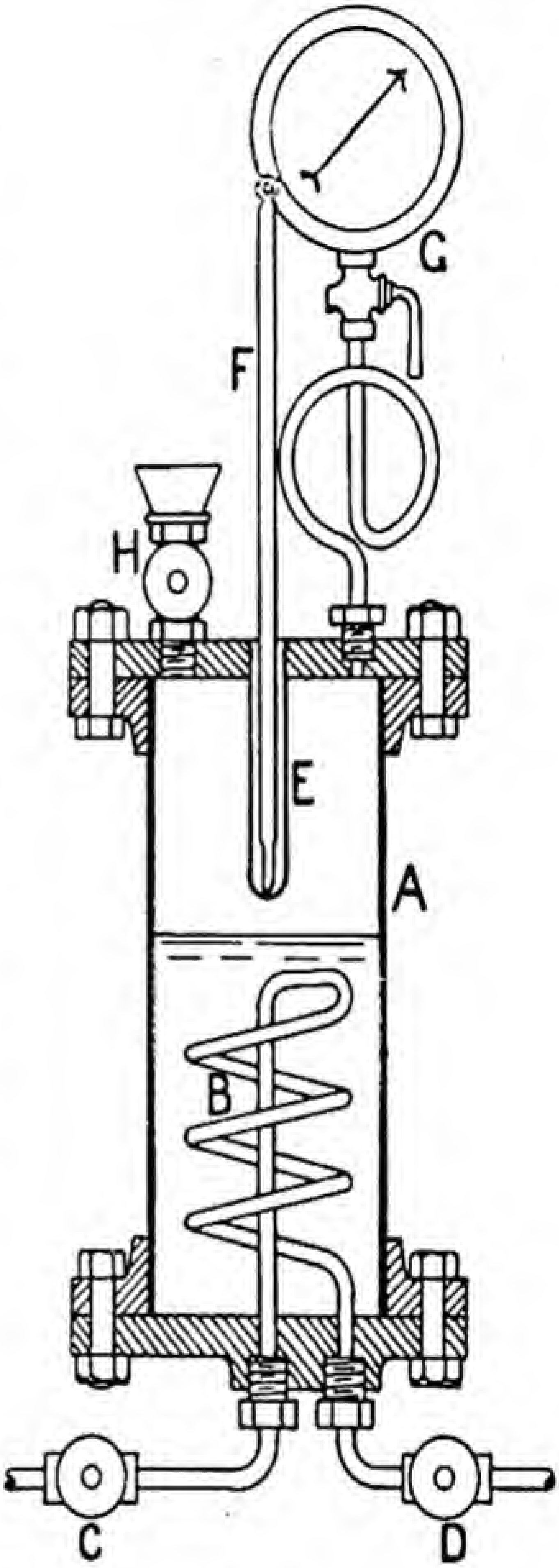


FIG. 50.—Apparatus for determining the relation of p and t for saturated steam.

Open valves *C* and *D* slightly, when the water in the vessel will be quickly heated by the heat coming from the steam condensed in the coil. The valve *H* ought to be left full open so as to get rid of air.

When steam is being freely discharged through H, close H, when the pressure of the steam will rise as shown by the gauge G, the temperature for different pressures being indicated by the thermometer F. It will be found impossible to obtain a pressure of steam in this apparatus higher than that in the steam boiler, hence the advantage over the ordinary experimental boiler heated by gas jets, in which dangerous pressures may be attained through careless use and safety valves which stick. The apparatus may be used for pressures up to 100 lbs. per square inch.

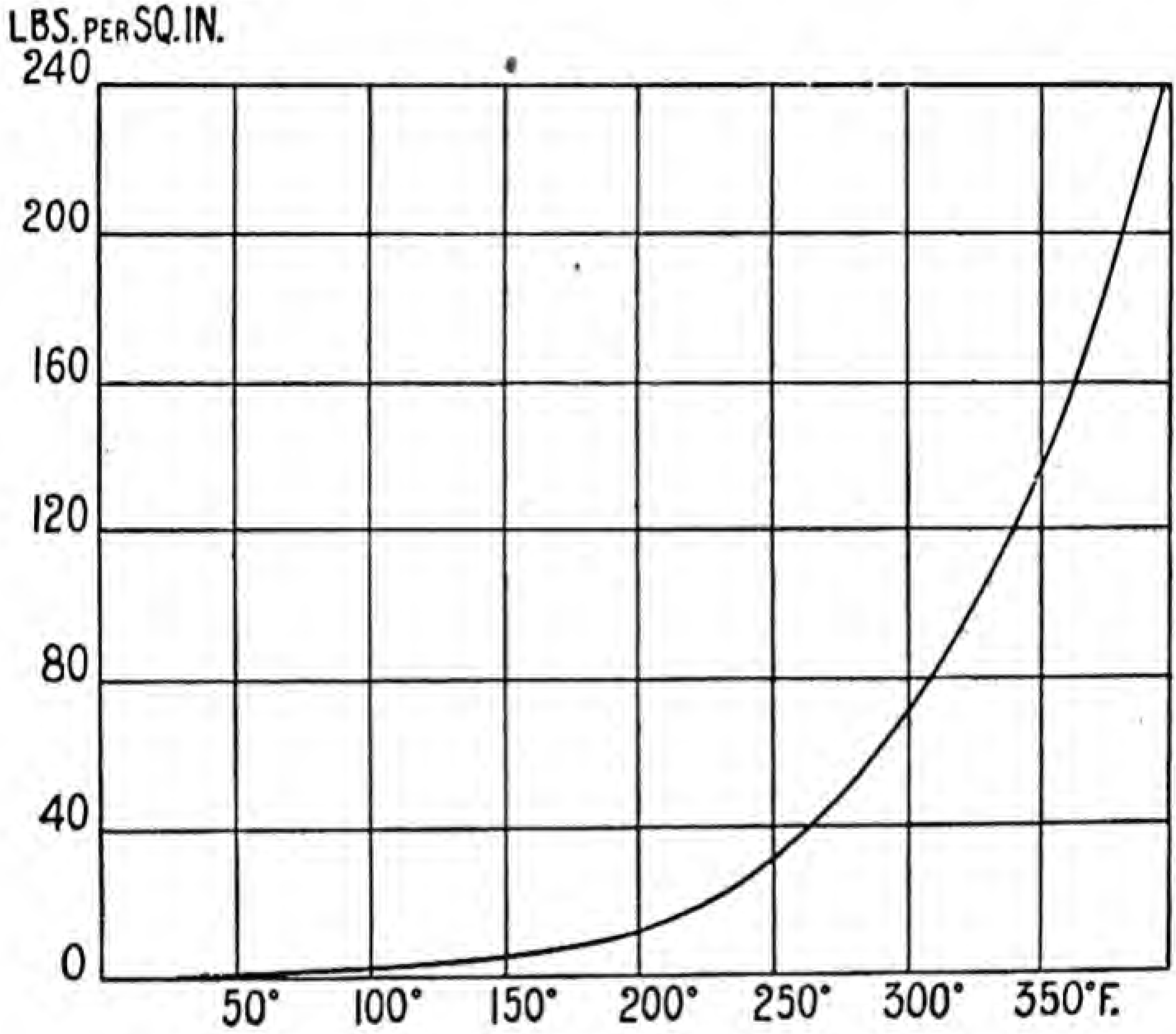


FIG. 51.—Curve showing the relation of p and t for saturated steam.

Read pressures and corresponding temperatures throughout the possible range and tabulate:

Pressure lbs. per square inch.	Temperature from experiment.	Temperature from Table (p. 454).	Error.

Plot columns 1 and 2, and also 1 and 3, and compare the resulting curves.

Pressure and volume of saturated steam.—This relation again does not follow any simple law. Various empirical formulae have been devised to show the relation, two of which are here quoted:

Rankine's formula, $pV^{1\frac{1}{16}}=479$;

Zeuner's formula, $pV^{1.0646}=479$.

Where p = pressure in lbs. per square inch,

V = volume in cubic feet per pound weight of steam.

The formulae differ only in the index of the power to which V is raised.

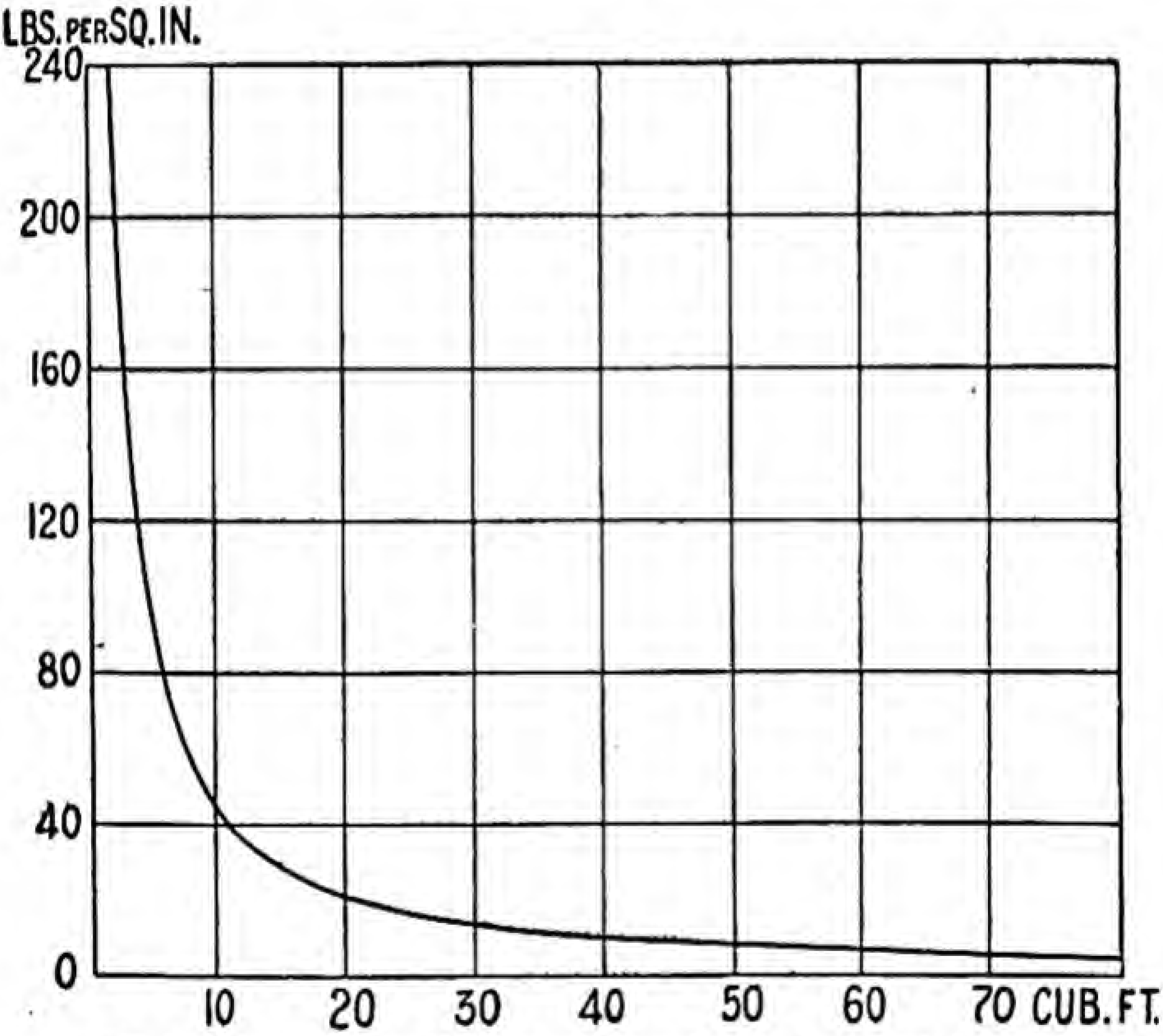


FIG. 52.—Curve showing the relation of p and v for dry saturated steam.

Direct experiments on the volume occupied by one pound of saturated steam at a given pressure are not easily performed, and can never be said to be strictly accurate. Plotting pressure and volume from the data in the Table (p. 454), as shown in Fig. 52, it will be seen that the curve somewhat resembles that representing Boyle's law (Fig. 41).

The student should now plot, on a large sheet of squared paper,

p and t, and also p and v, both curves being drawn on the same sheet and preserved for future reference.

Latent heat of steam.—One pound of steam possesses a latent heat of 967 B.T.U. only when formed at a pressure of one atmosphere. For each degree Fahrenheit of increase in temperature above 212° F. at which boiling occurs, it is found that the latent heat is diminished by 0·695 B.T.U., and is increased by the same amount for each degree Fahrenheit below 212° F.

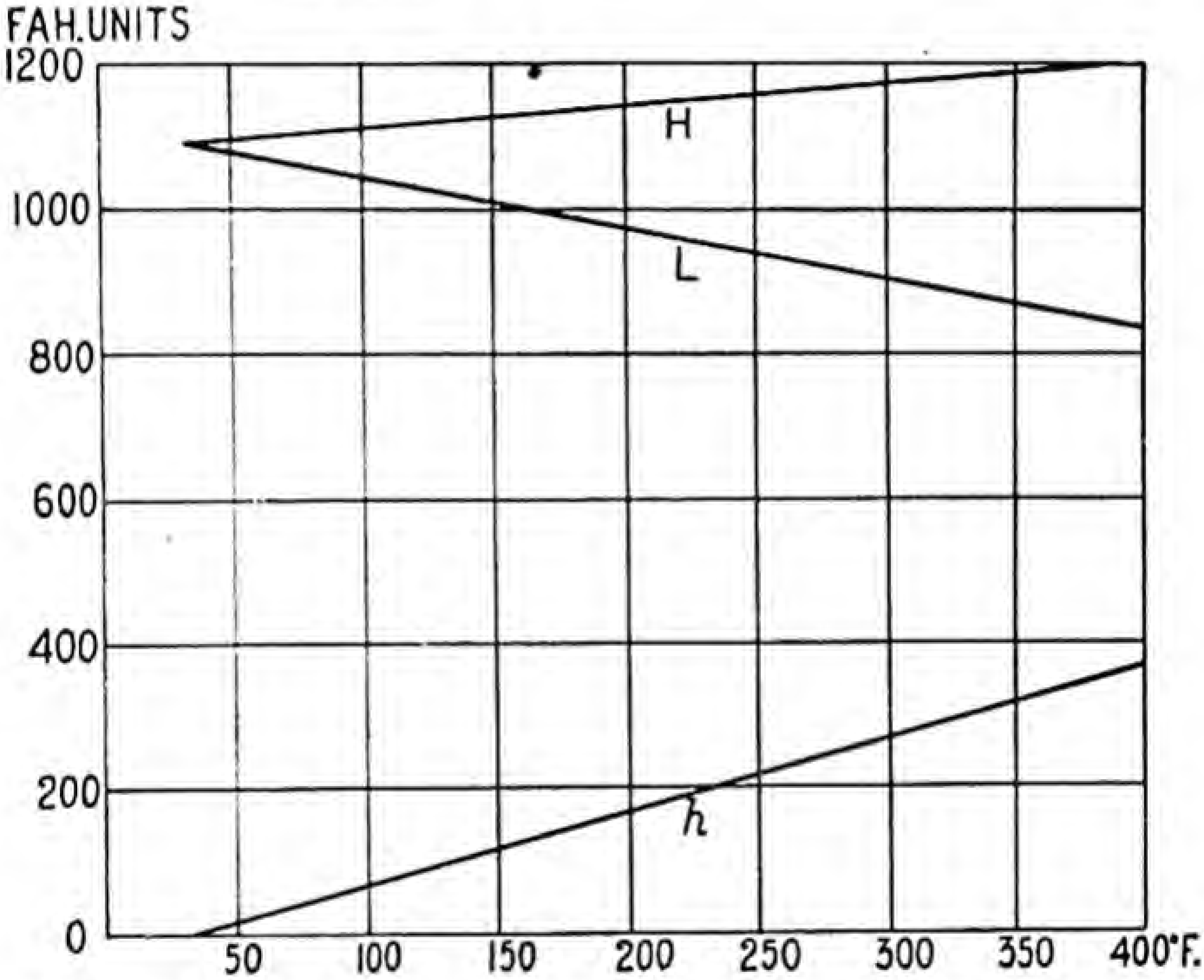


FIG. 53.—Curves showing H, L and h with temperature for dry saturated steam.

A formula to represent this would be

$$\begin{aligned} L_t &= 967 - 0{\cdot}695(t^\circ \text{ F.} - 212) \\ &= 967 - 0{\cdot}695t^\circ \text{ F.} + 147 \\ &= 1114 - 0{\cdot}695t^\circ \text{ F.}, \end{aligned}$$

where L_t = latent heat in B.T.U. of the pound of steam formed at t° F.

The formula may be modified to suit other scales of temperature. Thus: $L_t = 606{\cdot}5 - 0{\cdot}695t^\circ$ C. in which the latent heat is stated in lb.-degree-Cent. units of heat.

Total heat of steam.—Regnault's total heat of steam is defined as **the total heat** (sensible and latent), **which must be imparted to**

one pound of water at freezing temperature in order to convert it into saturated steam at any given temperature.

Let H = total heat of steam.
h = sensible heat.
L = latent heat.

Then $H = h + L$.

Putting $L = (1114 - 0{\cdot}695t^\circ\,\text{F.})$.
$h = (t^\circ\,\text{F.} - 32^\circ)$.
$H = (t^\circ\,\text{F.} - 32^\circ) + (1114 - 0{\cdot}695t^\circ\,\text{F.}) = 1082 + 0{\cdot}305t^\circ\,\text{F.}$

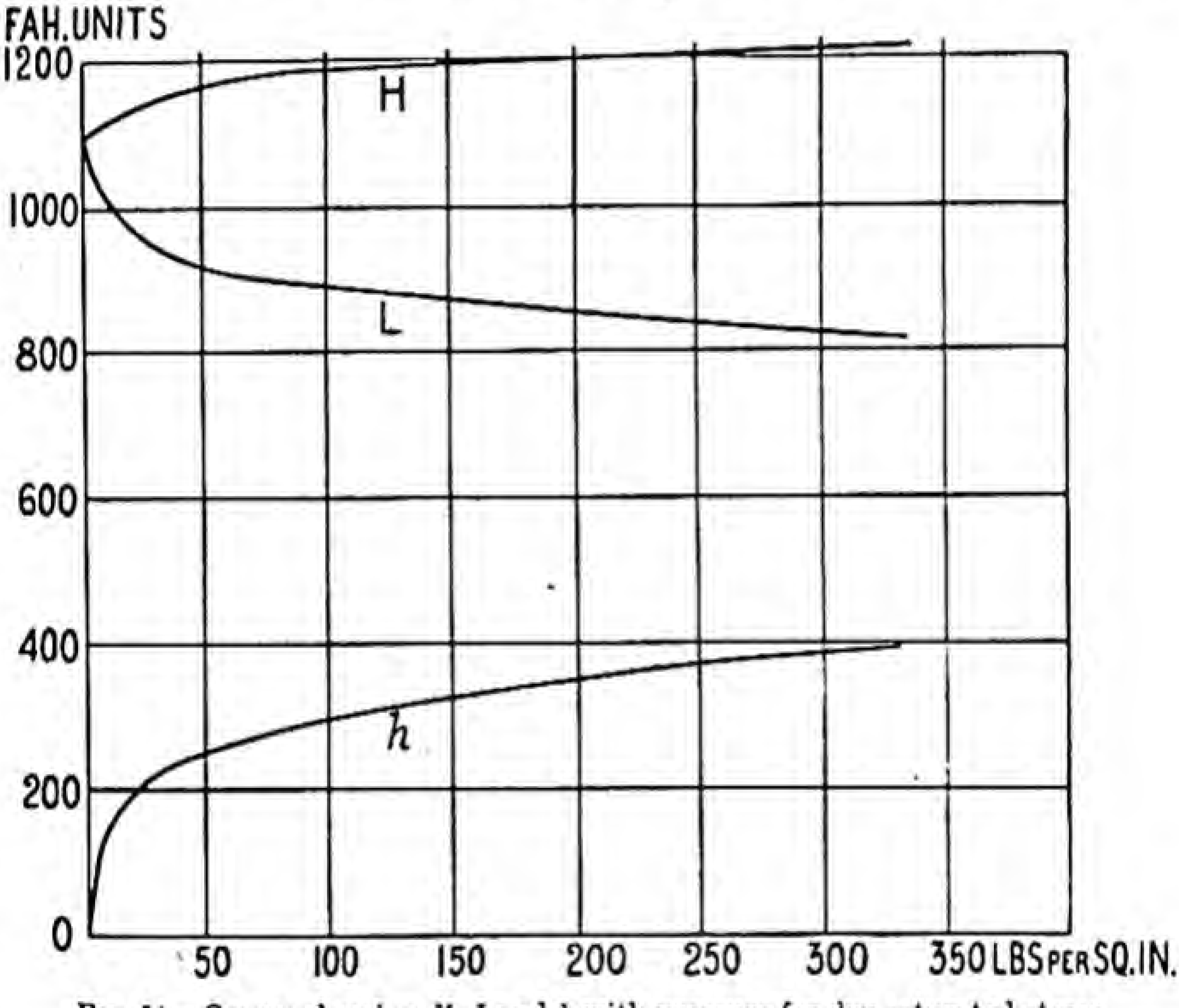


Fig. 54.—Curves showing H, L and h with pressure for dry saturated steam.

This equation may be used to calculate values of H. It may be modified to express the result in lb.-degree-Cent. units, thus giving

$$H = 606{\cdot}5 + 0{\cdot}305t^\circ\,\text{C.}$$

Values of H, L and h will be found in the Table, p. 454. Taking these values, plot H, L, h and temperatures; also plot the same quantities and pressures, putting each set of three curves on a large sheet of squared paper. The curves will resemble those shown in Figs. 53 and 54, and should be preserved for future reference.

It should be noted that H, L and h give straight lines when plotted with temperatures, and that the value of H throughout the

range alters very little comparatively. When plotted with pressures it will be noticed that all three change rapidly at low pressures and tend to take up a more uniform rate of change as the pressure increases.

Limitations of steam tables.—Steam tables usually terminate at a pressure of 300 lbs. per square inch. Tables which go beyond a pressure of 350 lbs. per square inch, corresponding to a temperature of 432° F., are not trustworthy if the quantities beyond these limits have been calculated from empirical formulae such as those given above.

Water expands on being heated, and, as the temperature at which the generation of steam begins is increased, the volume of steam from a given weight of water diminishes. It follows, therefore, that a point will be reached where the volume of the steam will be equal to the volume of the water; at this point steam and water cannot be distinguished from each other, and the latent heat becomes zero. The temperature at which this occurs lies in the region of from 365° to 370° C., and is known as the **critical temperature.**

The subject of high steam pressures is becoming of increasing importance. De Laval has already used steam at a pressure of from 1500 to 1700 lbs. per square inch in his steam turbines, and has experimented with even higher pressure steam. Such pressures are not likely to be employed in the ordinary reciprocating engine. Some results obtained by the Laval Steam Turbine Co.[1] are here given:

Temp. Fah. degrees.	Pressure lbs. per sq. inch.
417·2	284·5
497·2	711·2
593·6	1422·3
651·2	2133·5
698·0	2844·6

There is room for a great amount of further experimental work in connection with the properties of high pressure steam.

[1] See Article, "Steam at high pressures," *Engineering*, Jan. 4th and 11th, 1907.

Heat required to form superheated steam.—This amount of heat may be roughly calculated on the assumption that the specific heat of superheated steam is constant, and equal to 0·48. Proceed thus: Calculate first the total heat of saturated steam at the given pressure from

$$H = 1082 + 0{\cdot}305t^{\circ}\text{F}.$$

t° being the temperature of boiling corresponding to the given pressure.

Let t_s = the temperature Fahrenheit at which it is desired the steam to be.

Heat required to superheat the steam from t° F. to t_s° F. will be given by

$$0{\cdot}48(t_s - t).$$

$$\therefore \text{ Total heat required in B.T.U.} = H + 0{\cdot}48(t_s - t)$$
$$= 1082 + 0{\cdot}305t^{\circ} + 0{\cdot}48(t_s - t)$$
$$= 1082 - 0{\cdot}175t^{\circ} + 0{\cdot}48t_s.$$

Captain H. Riall Sankey[1] states that for practical purposes the specific heat of steam at constant pressure may be taken as 0·6, and at constant volume 0·46, and uses these numbers as being the most trustworthy yet obtained. Using 0·6 instead of the older number given above, we obtain for the total heat required to produce superheated steam at constant pressure

$$H + 0{\cdot}6(t_s - t).$$

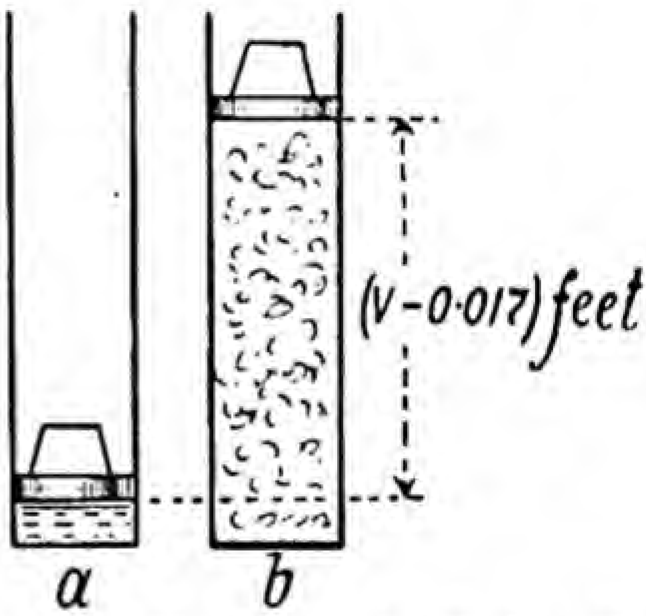


FIG. 55.—Diagram showing the formation of steam at constant pressure.

Formation of steam at constant pressure.—Fig. 55, *a*, shows a cylinder fitted with a piston which may be loaded to any desired extent. One pound of water is contained in the cylinder, the piston exerting a constant pressure p lbs. per square inch on it. Let the area of the piston be 1 square foot. Since the volume of 1 lb. of water is 0·017 cubic foot, the length of cylinder occupied by the water will be 0·017 foot. Suppose that the water is already at the boiling temperature corresponding to p, and that the latent heat required to form steam at this temperature

[1] See *The Energy Chart*, Capt. Sankey (Frost & Sons, Rugby).

is supplied to it. The volume will thereby be increased to V cubic feet = volume occupied by one pound of steam at the given pressure p; the piston will rise to accommodate this increased volume, which will evidently occupy a length V feet of the cylinder (Fig. 55, *b*).

Two things have occurred in this process, (*a*) the state has been changed from liquid to gaseous, (*b*) external work has been done against the resistance p, and both are due to the latent heat supplied. The external work done may be easily calculated.

Total pressure on piston $= 144p$ lbs.

Distance through which piston is moved $= (V - 0{\cdot}017)$ feet.

External work done $= 144p\,(V - 0{\cdot}017)$ foot-lbs.

It is usually sufficiently accurate to neglect the volume of one pound of water, and doing this, we may write

$$\text{External work done} = 144pV \text{ foot-lbs.}$$
$$= \frac{144pV}{J} \text{ heat units.}$$

This expression gives the portion of the latent heat supplied which has been transformed into mechanical work. The remainder of the heat energy supplied, viz.

$$\left(L - \frac{144pV}{J}\right) \text{ heat units}$$

remains as internal energy in the steam. The **total internal energy, or intrinsic energy** of the steam is defined as **the total heat of formation H diminished by the heat required to perform external work.**

Writing I for the intrinsic energy,

$$I = H - \frac{144pV}{J} \text{ heat units.}$$

EXERCISES ON CHAPTER V.

1. Explain the terms "sensible heat," "latent heat," and "total heat of steam."

2. Using the Table, p. 454, plot pressure and temperature of saturated steam. State in general terms how the pressure varies with the temperature.

3. Plot pressure and volume of dry saturated steam, using the Table, p. 454. Beginning at a point near the top of the curve, plot on

the same sheet a curve to represent Boyle's law. Mark the curves clearly in the diagram and contrast them.

4. Using the empirical formula

$$\log_{10} p = 6{\cdot}1007 - \frac{B}{\tau} - \frac{C}{\tau^2},$$

calculate the pressure of saturated steam corresponding to a temperature of 300° F. Compare your result with the tabular result (p. 454).

5. Using the empirical formula

$$pV^{\frac{17}{16}} = 479,$$

calculate the volume occupied by one pound weight of dry saturated steam at a pressure of 100 lbs. per sq. inch absolute. Compare your answer with the tabular number, p. 454.

6. Calculate the latent and total heats of dry saturated steam at a temperature of 300° F. Use the empirical formulae

$$L_t = 1114 - 0{\cdot}695t^\circ \text{ F.}$$
$$H = 1082 + 0{\cdot}305t^\circ \text{ F.}$$

Compare your results with those given in the Table, p. 454.

7. Taking the specific heat of superheated steam to be 0·6 at constant pressure, how much heat must be supplied per hour to dry saturated steam at 100 lbs. per square inch absolute in order to superheat it to 450° F.? The weight of steam to be treated per hour is 2000 lbs.

8. Distinguish between the internal and external work done during the formation of dry saturated steam at constant pressure.

9. Taking the quantities required from the Table, p. 454, calculate how much heat must be abstracted from 100 lbs. weight of dry saturated steam at 3 lbs. per square inch absolute in order to condense and cool it to 130° F.

10. The water in a tank containing 40 gallons is heated by blowing dry saturated steam into the water at a pressure of 30 lbs. per square inch absolute. Suppose the initial temperature of the water to be 60° F., calculate what weight of steam must be used to raise the temperature to 150° F. Take any quantities you require from the Table, p. 454.

11. What heat must be given to 1 lb. of water at 80° F. to convert it into steam at 300° F.? Regnault's formula for the *total heat* of a pound of steam from water at 32° F. being $H = 1082 + 0{\cdot}305t$ where t° F. is the temperature of the steam, how many pounds of this steam are equivalent in *total heat* to the calorific power (15,000 units of heat) of a pound of coal? 1898.

12. A formula for Regnault's total heat H will be found in the Tables, p. 452; it is the total heat which must be given to 1 lb. of water at 0° C. to raise its temperature as water to t° C., and then to convert it all into steam at t° C. What is the heat which must be given to 1 lb. of water at 40° C. to convert it into steam at 170° C.? 1906.

13. With a small experimental boiler you are finding the pressure of steam when its temperature is, say, 100° C., 110° C., 120° C., etc. Show, with sketches, exactly how you would proceed. In what way does the presence of air with the steam spoil your results? **1906.**

CHAPTER VI.

THE DIAGRAM OF WORK.

Work.—A force is said to be doing **work** when it acts through a distance, overcoming resistance. Work is measured by the product of the magnitude of the force and the distance through which it acts, the latter being measured along, or parallel to, the line of action of the force. The **unit of work** generally used in this country is the **foot-pound, and is that quantity of work which is done when a force of one pound acts through a distance of one foot in its line of action.**

Other units occasionally used are the **inch-ton** and the **foot-ton**; these are sufficiently defined by their names.

In the metric system, the **erg** is the unit of work, and is performed when a force of one dyne acts through a distance of one centimetre. As this represents a very small quantity of work, engineers find it more convenient to use the **metre-kilogram**, which is the work done when a force equal to the weight of one kilogram acts through a distance of one metre.

A foot-pound of work is equivalent to $1{\cdot}3562 \times 10^7$ ergs. A metre-kilogram is equivalent to $7{\cdot}235$ foot-pounds.

EXAMPLE. Supposing a uniform force of 10,200 lbs. to act on the piston of a steam engine, the stroke of which is 2 feet, calculate the work done per stroke.

$$\begin{aligned} \text{Work done} &= \text{force} \times \text{distance} \\ &= 10{,}200 \times 2 \\ &= \underline{20{,}400} \text{ foot-lbs.} \end{aligned}$$

Power.—**Power** is the name given to the rate of performing work. The **unit of power** used generally in this country is the

horse-power, and is the rate of doing work such that 33,000 foot-lbs. are performed in one minute. This unit was defined by James Watt; it is really about 50 per cent. greater than the rate of working of an average horse. The horse-power developed in any given case is ascertained by first calculating the work done per minute, in foot-pounds, and then dividing the result by 33,000.

EXAMPLE. In the last example (p. 75), the piston makes 240 strokes per minute. Calculate the horse-power.

$$\text{Work done per stroke} = 20{,}400 \text{ foot-lbs.}$$
$$\text{Work done per minute} = 20{,}400 \times 240$$
$$= 4{,}896{,}000 \text{ foot-lbs.}$$
$$\text{Horse power} = \frac{4{,}896{,}000}{33{,}000}$$
$$= \underline{148 \cdot 4.}$$

Graphic representation of work.—Since work is measured by the product of two quantities, force and distance, we may represent

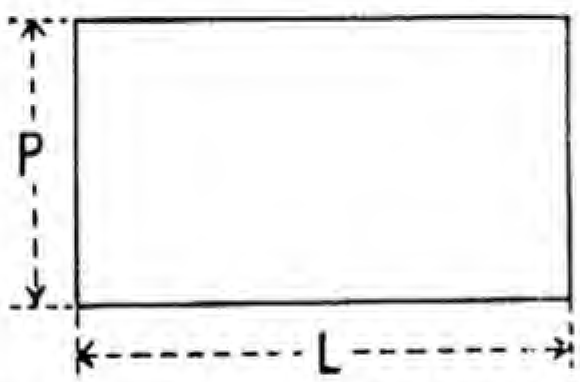


FIG. 56.—Diagram of work done by a uniform force.

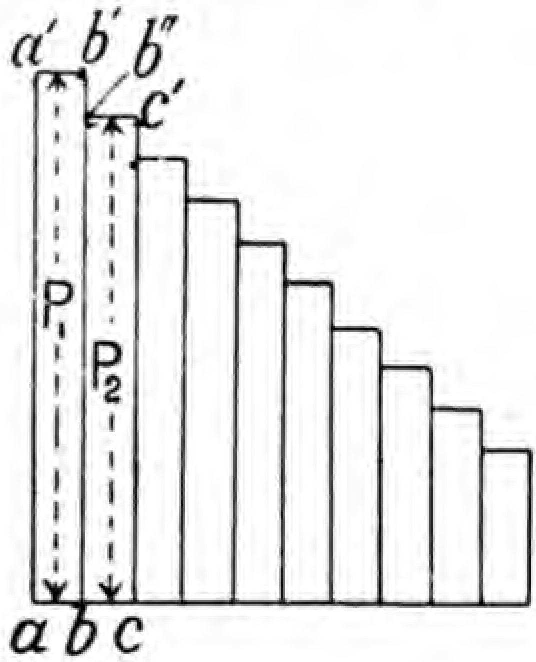


FIG. 57.—Diagram of work done by a force varying in steps.

it by an area. Thus, supposing a uniform force P lbs. to act on the piston of an engine, the stroke of which is L feet, the work done will be $P \times L$ foot-lbs. Set off L to scale in a diagram (Fig. 56), and erect ordinates to represent P to some convenient scale. These ordinates will be of constant height as P is supposed to be uniform. The result is a rectangular diagram of area equal to $P \times L$, which therefore represents the work done in this special case of steam when used non-expansively.

In the case of a varying force, we may first imagine the force to remain constant in magnitude for short distances. Thus, in Fig. 57, P_1 P_2, etc., are supposed to be uniform over the short distances ab, bc, etc., into which the length of the diagram is divided

Work done by $\quad P_1 = P_1 \times ab$;

" " $\quad P_2 = P_2 \times bc$, etc.

Now $\quad P_1 \times ab$ = area of the rectangle $abb'a'$,

and $\quad P_2 \times bc$ = area of the rectangle $bcc'b''$, etc.

Hence the total work done is represented by the sum of the areas of the rectangles, *i.e.* by the area of the whole figure.

Had the intervals ab, bc, etc., been taken smaller, the principle would still be true, even in the special case of the intervals being so small that the steps in the upper portion of the diagram become changed into a continuous curve as in Fig. 58. In this case, also, the work done will be represented by the area of the diagram, which may be found by taking the product of its average height and its length. The result may be stated in foot-pounds by measuring the average height by the scale of force in pounds and the length L by the scale of feet. The average height so measured may be called the average or mean force, and writing this as P_m in pounds we obtain—

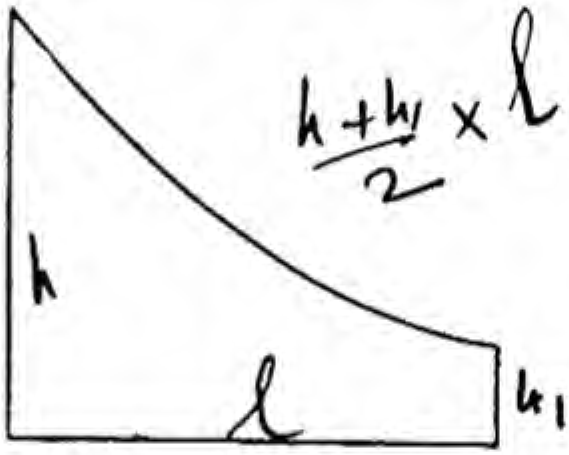


FIG. 58.—Diagram of work done by a continuously varying force.

$$\text{Work done by a varying force} = P_m \times L \text{ foot-lbs.}$$

Diagram of work done on the piston.—In modern steam engines, the steam is used **expansively** in almost all cases. The steam is admitted freely to the cylinder during a portion of the stroke of the piston, and is then cut off by the valve closing the port, the remainder of the stroke being completed under the action of the continually diminishing pressure of the steam as it expands. A diagram may be drawn of the work done on the piston, which, while it will not show exactly what is occurring, will be found to be useful for comparison with the true diagram to be described later. For this purpose it may be assumed that the steam has

uniform pressure during the admission period, and that after "cut off" the expansion follows the equation which represents Boyle's law for perfect gases, viz.:

$$pv=c.$$

It may also be assumed that the valves open and close very quickly so as to make a sharp difference between admission and expansion, expansion and exhaust, etc. An example will make the method clear.

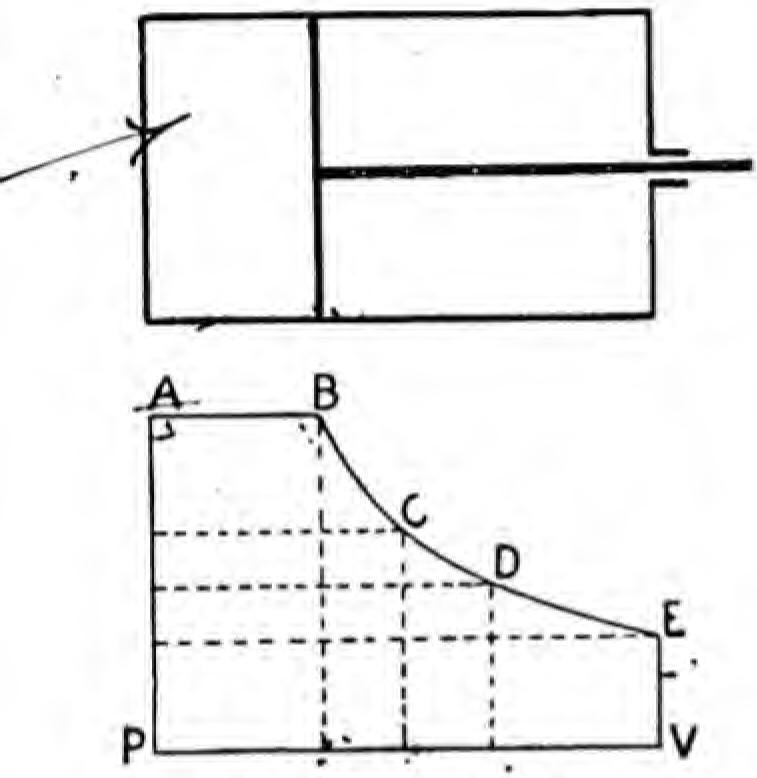


FIG. 59.—Diagram of work done in a steam engine cylinder.

EXAMPLE. An engine has a piston 10″ diameter and 18″ stroke. Steam is admitted during one-third of the stroke at a pressure of 60 lbs. per sq. inch absolute, and is then cut off, the remainder of the stroke being accomplished under the expansive action of the steam. Draw a diagram showing the work done on one side of the piston.

Draw PV (Fig. 59) of length to represent to scale the stroke of 18″ and take this to be the perfect vacuum line, *i.e.* the line showing zero pressure. Draw PA perpendicular to PV and make it of height to represent a pressure of 60 lbs. per square inch to scale. Draw AB parallel to PV, making AB equal to one-third of PV. AB will then represent the admission of steam at constant pressure during one-third of the stroke. We now proceed to find points on the expansion curve.

The volume of steam in the cylinder at B will be found from

$$v_1=\text{area of piston} \times AB.$$

The volume at any other point in the stroke will be

$$v=\text{area of piston} \times \text{distance travelled by piston up to that point.}$$

Since the expression for every volume contains the factor "area of piston," this may be omitted, and the volume simply taken as proportional to the distance travelled by the piston. Thus we may take the numbers 6, 9, 12 and 18 as representing the volumes at $\frac{1}{3}$, $\frac{1}{2}$, $\frac{2}{3}$ and end of the stroke, shown by B, C, D and E respectively in Fig. 59.

Application of the law $pv = c$

gives $p_1 v_1 = p_2 v_2$.

At $\frac{1}{2}$ stroke, $p_2 = \frac{p_1 v_1}{v_2} = \frac{60 \times 6}{9} = 40$ lbs. per sq. inch at C.

At $\frac{2}{3}$ stroke, $p_3 = \frac{p_1 v_1}{v_3} = \frac{60 \times 6}{12} = 30$ lbs. per sq. inch at D.

At end of stroke, $p_4 = \frac{p_1 v_1}{v_4} = \frac{60 \times 6}{18} = 20$ lbs. per sq. inch at E.

Plotting these pressures, B, C, D and E will be found, and an even curve drawn through these will give the expansion curve.

Since the ordinates in the now completed diagram of work $PABEV$ represent pressures per square inch, the area of the diagram will give the work done on the piston per square inch of piston area; the total work may be calculated by multiplying the work done per square inch by the area of the piston in square inches.

Net work done on the piston.—The work done by the steam on one side of the piston is expended partly in overcoming the resistance offered by the piston and partly against the back pressure exerted on the other side of the piston by the exhaust steam. In the case of a non-condensing engine, discharging its steam into the atmosphere, the back pressure would be about 17 or 18 lbs. per square inch absolute; a condensing engine would have a back pressure of 2 or 3 lbs. per square inch absolute. For our present purpose, we may suppose these back pressures to be maintained uniform throughout the stroke.

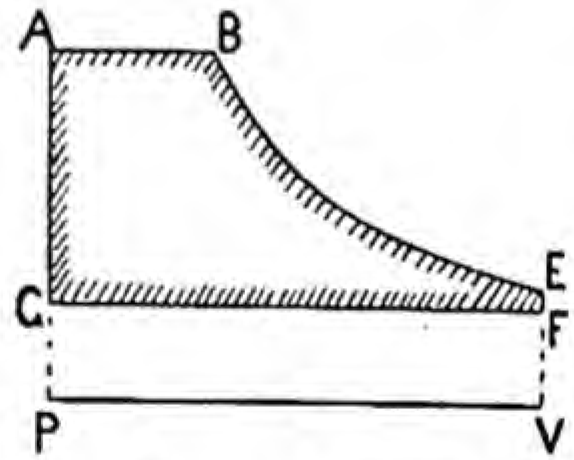


Fig. 60.—Diagram showing net work done on the piston.

The net work done on the piston per square inch of its area will be the difference between the area of the diagram of work $PABEV$ (Fig. 59), and the work done against the back pressure. In Fig. 60, let PG represent the back pressure to scale. Draw GF parallel to PV; then, since the back pressure is supposed uniform, the area $PGFV$ will represent the work done in overcoming the back pressure. The area $GABEF$ will now represent the net work done on the piston per square inch of its area.

Calculation of net work done on the piston.—This may be effected by the following method which is in common use by engineers. The average pressure on the piston is given by multiplying the average height of the diagram *GABEF* by the scale of pressure. Divide *GF* into ten equal parts and erect an

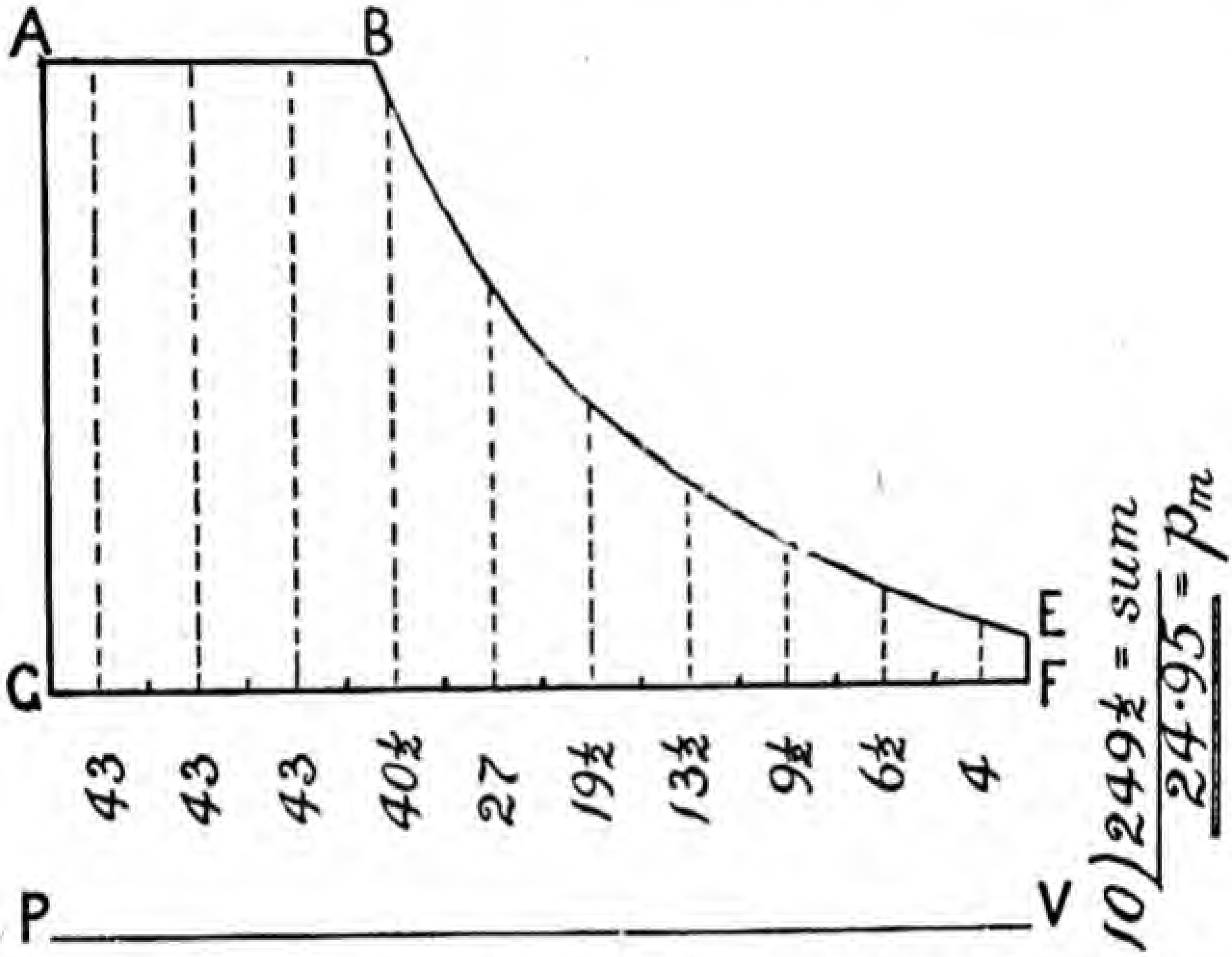


FIG. 61.—Diagram showing a method of estimating the mean pressure.

ordinate at the middle of each part (Fig. 61). Measure the heights of these ordinates, take their sum, and divide by 10. The heights have been measured in Fig. 61 by use of a scale of pressures.

Let p_m = average pressure, as found from the diagram, in lbs. per square inch.

L = length of stroke in feet.

A = area of piston, square inches.

Then

$$\text{Work done per stroke} = p_m \times A \times L \text{ foot-lbs.}$$

Fig. 61 shows the measurements and calculations giving p_m for data as in the example, p. 78, a back pressure of 17 lbs. per square inch being assumed. Hence, in the example given

$$\text{Work done per stroke} = 24{\cdot}95 \times 78{\cdot}54 \times 1{\cdot}5$$
$$= \underline{2939} \text{ foot-lbs.}$$

Equation for mean pressure.—In place of determining the mean pressure from a diagram drawn carefully to scale as shown above, engineers often use the equation

$$p_m = p_1\left(\frac{1+\log_e r}{r}\right) - p_b.$$

In this equation

p_1=absolute pressure of steam supplied, in lbs. per square inch,

p_b=the absolute back pressure, in lbs. per square inch,

p_m=the net average pressure, in lbs. per square inch.

The quantity denoted by r is called the **ratio of expansion**, and is calculated by **dividing the volume of steam in the cylinder at the end of the stroke by the volume of steam in the cylinder at the point of cut off.** The quantity $\log_e r$ is the **hyperbolic logarithm** of the ratio of expansion; values of these logarithms are given in the Table on p. 453.

EXAMPLE. Let

p_1=60 lbs. per square inch absolute,

p_b=17 ,, ,, ,, ,,

r= 3

$\log_e r$= 1·0986 (from the Table).

Then

$$p_m = p_1\left(\frac{1+\log_e r}{r}\right) - p_b$$

$$=60\left(\frac{1+1{\cdot}0986}{3}\right) - 17$$

$$=24{\cdot}97 \text{ lbs. per square inch.}$$

For the same data, the result found from the diagram is 24·95, as shown in Fig. 61. Owing to unavoidable inaccuracies in drawing and measurement, the result as obtained from the diagram differs slightly from the more accurate result found from the equation.

The diagram in Fig. 61 has been constructed with the omission of several considerations which modify its shape. Such a diagram is never obtained in practice from any steam engine. Engineers give the name **hypothetical diagram** to Fig. 61 to distinguish it from the more correct diagram which we now proceed to discuss.

Reference is made to Fig. 62, in which the hypothetical parts of the diagram are shown dotted.

Admission.—It is customary to arrange the valve in such a manner that steam is admitted to the cylinder a little before the end of the exhaust stroke. The object of doing so is to provide a plentiful supply of steam in the cylinder when the piston is ready

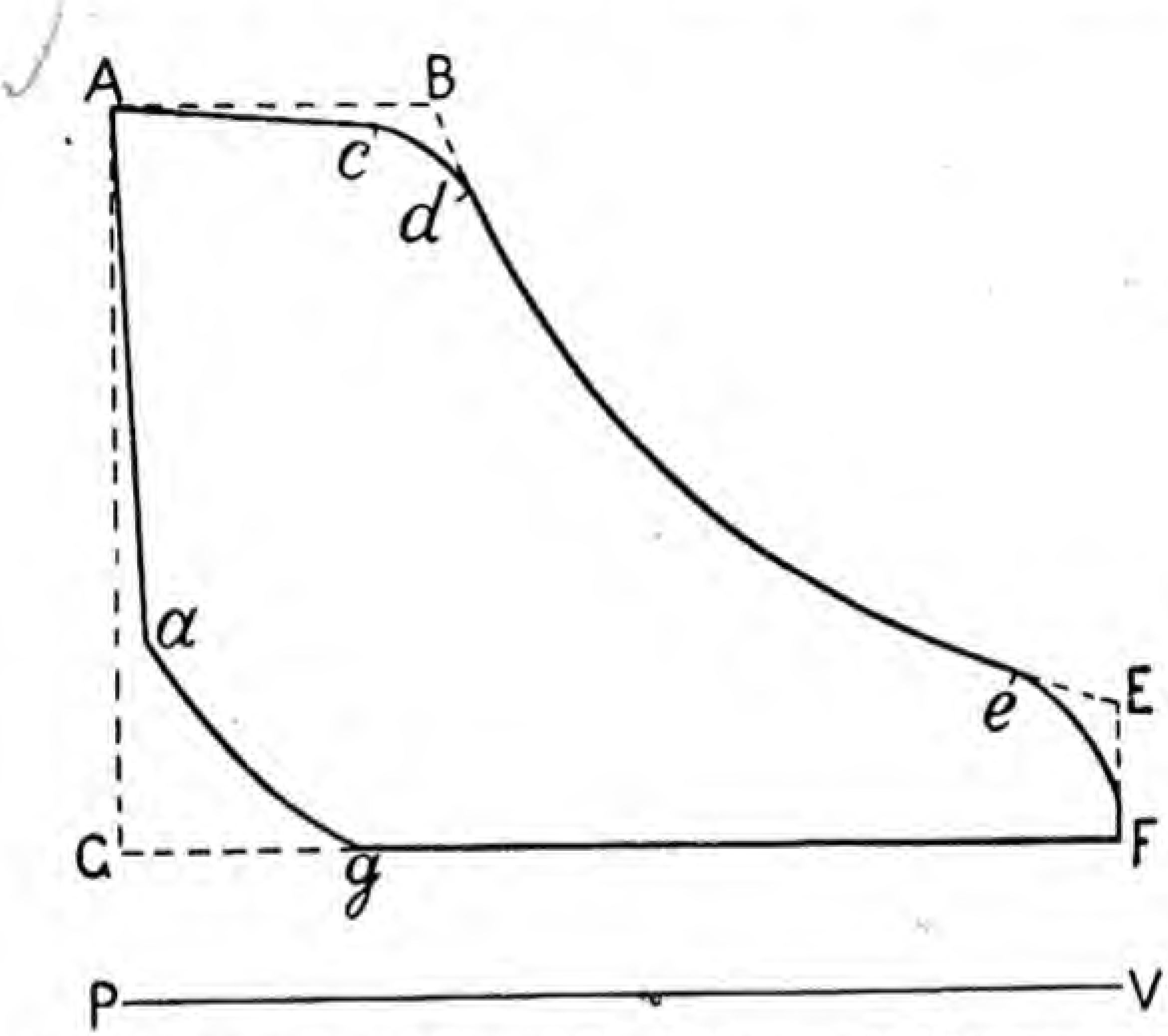


Fig. 62.—Comparison of the probable and hypothetical diagrams of work.

to begin the stroke. This may produce a line *aA*, not quite vertical.

Throttling and wiredrawing.—As the piston moves along the cylinder from the commencement of the stroke, it gradually increases its speed. The rate at which the steam must flow into the cylinder will therefore continually increase, and, if the passages are restricted in area, the steam may be **throttled**, *i.e.* not able to flow into the cylinder quickly enough to follow up the piston, with a consequent gradual fall of pressure. The effect is shown by the gradually falling line *Ac* in Fig. 62. Further, the valve always closes more or less slowly, with the result of increasing the drop in pressure near the corner *B*. The steam near the point of cut off has to pass through the very small and continually diminishing opening left by the valve and is said to be **wiredrawn.** The effect

is shown by the rounded corner *cd* in Fig. 62. To produce a sharp corner at *B* would require an instantaneous closing of the valve from full open to full shut.

Expansion.—The actual expansion law followed by the steam is very complex owing to the action of the cylinder walls, which are alternately hot and cool. The effect is to produce a varying quantity of water in the cylinder. In general, however, the actual expansion curve follows very closely the law

$$pv = \text{a constant.}$$

The actual conditions will be more fully discussed later.

Exhaust.—It is found best to have the exhaust valve arranged so as to open before the end of the stroke. This allows the steam to get away quickly through the exhaust passages and so tends to reduce the back pressure. Directly the exhaust valve opens at the point *e* in Fig. 62 the pressure begins to fall, producing a rounded corner *eF*. During the exhaust stroke the pressure may or may not remain uniform, being affected by the area of the exhaust passages, which, if too small, will cause the back pressure to rise near the middle of the stroke where the piston is moving rapidly.

Cushioning.—It is customary to close the exhaust valve at a point considerably before the end of the exhaust stroke is reached, thus entrapping some of the exhaust steam in the cylinder. The piston returning to the end of the stroke compresses this steam, which thus serves as a soft **cushion** for bringing the piston to rest at the end of the stroke. Cushioning is rendered possible by the presence of **clearance volume.** This volume is defined as the volume behind the piston when it is at the beginning of the stroke, and includes any space left in the cylinder between the piston and the cylinder cover for the purpose of enabling the piston to work without coming against the cover, and also the volume of the passages (as far as the valves) by which the steam is admitted and exhausted. If there were no clearance volume it is obvious that there could be no cushioning, as it is impossible to compress steam or any other substance to such a degree that its volume becomes zero.

Cushioning also serves the purpose of increasing the pressure in the clearance volume to a value considerably above the exhaust pressure before fresh steam is admitted. The effect is to produce

more economical working, as less steam will require to be admitted in order to bring the pressure in the clearance volume up to steam chest pressure at the commencement of the stroke.

In Fig. 62, cushioning starts at *g*, at which point the exhaust valve closes, and the gradual rise in pressure as the compression goes on is shown by the curve *ga*.

The general effect of all the points enumerated is to reduce the area of the diagram of work by an amount which varies in engines of different classes, amounting to from 25 to 40 per cent. The actual mean pressure which may therefore be obtained in practice will be from 75 to 60 per cent. of that obtained by calculation from the equation on p. 81.

Other effects of clearance.—Clearance also affects the volume of steam present in the cylinder at any point in the stroke. To obtain this volume, the clearance volume must be added to the volume swept by the piston from the commencement of the stroke up to that point. Further, the true ratio of expansion should be calculated from

True ratio of expansion

$$= \frac{\text{volume present in cylinder at end of stroke}}{\text{volume present in cylinder at point of cut off}}$$

$$= \frac{(\text{total volume swept by piston})+(\text{clearance volume})}{(\text{volume swept by piston up to cut off})+(\text{clearance volume})}.$$

In the example worked out on p. 81, the ratio of expansion was taken as the reciprocal of the fraction of the stroke at which cut off occurred, viz. 3. While this method is accurate enough for the purposes of the hypothetical diagram, the student should clearly understand that it gives an approximate result only.

Clearance is usually larger proportionally in small engines than in those of considerable size. Stating clearance as a percentage of the volume swept by the piston, we find it may range from about 20 per cent. in small engines to about 4 per cent. in large engines. The clearance volume may be calculated from the drawings of the cylinder, or it may be found experimentally by bringing the piston to the beginning of the stroke, closing the admission valve, and then ascertaining how much water is required to fill the space left. From the weight of this water, the volume may be easily calculated.

Example i. Supposing 4 lbs. of water are required to fill the clearance volume of a cylinder 10″ diameter × 18″ stroke of piston, calculate the clearance volume and express it as a percentage of the volume swept by the piston. Take the weight of one cubic foot of water to be 62·3 lbs.

Let v = clearance volume in cubic inches.

Then
$$v : 1728 = 4 : 62{\cdot}3,$$
$$v = \frac{1728 \times 4}{62{\cdot}3}$$
$$= \underline{111} \text{ cubic inches.}$$

$$\text{Volume swept by piston} = \frac{\pi d^2}{4} \times 18$$
$$= 78{\cdot}54 \times 18$$
$$= \underline{1414} \text{ cubic inches.}$$

$$\text{Percentage required} = \frac{111}{1414} \times 100$$
$$= \underline{7{\cdot}86.}$$

Example ii. Suppose in Example i. that the steam is cut off at one-third of the stroke, and calculate the true ratio of expansion.

$$\text{Volume present in cylinder at end of stroke} = 1414 + 111$$
$$= 1525 \text{ cubic inches.}$$

$$\text{Volume present in cylinder at cut off} = \frac{1414}{3} + 111$$
$$= 582{\cdot}3 \text{ cubic inches.}$$

$$\text{True ratio of expansion} = \frac{1525}{582{\cdot}3} = \underline{2{\cdot}62.}$$

Example iii. Use the particulars of Examples i. and ii. and represent the clearance volume in the diagram of work. Draw the admission and expansion parts of the diagram, taking the admission pressure to be 60 lbs. per square inch absolute.

In Fig. 63 let PV be the perfect vacuum line, and make PV to scale to represent the stroke of the piston. The length of PV multiplied by the area of the piston would give the volume swept by the piston, and we may therefore take PV as representing also the volume swept by the piston.

Calculate a length PO from

$$PO : PV = \text{clearance volume} : \text{volume swept by piston.}$$

PO will then represent the clearance volume to the same scale that PV represents the volume swept by the piston. The whole length OV will now represent the total volume present in the cylinder at the end of the stroke. Thus,

Let l=actual length of PO in inches, then

$$l : PV = 111 : 1414$$

$$\therefore\ l = \left(PV \times \frac{111}{1414}\right) \text{ inches.}$$

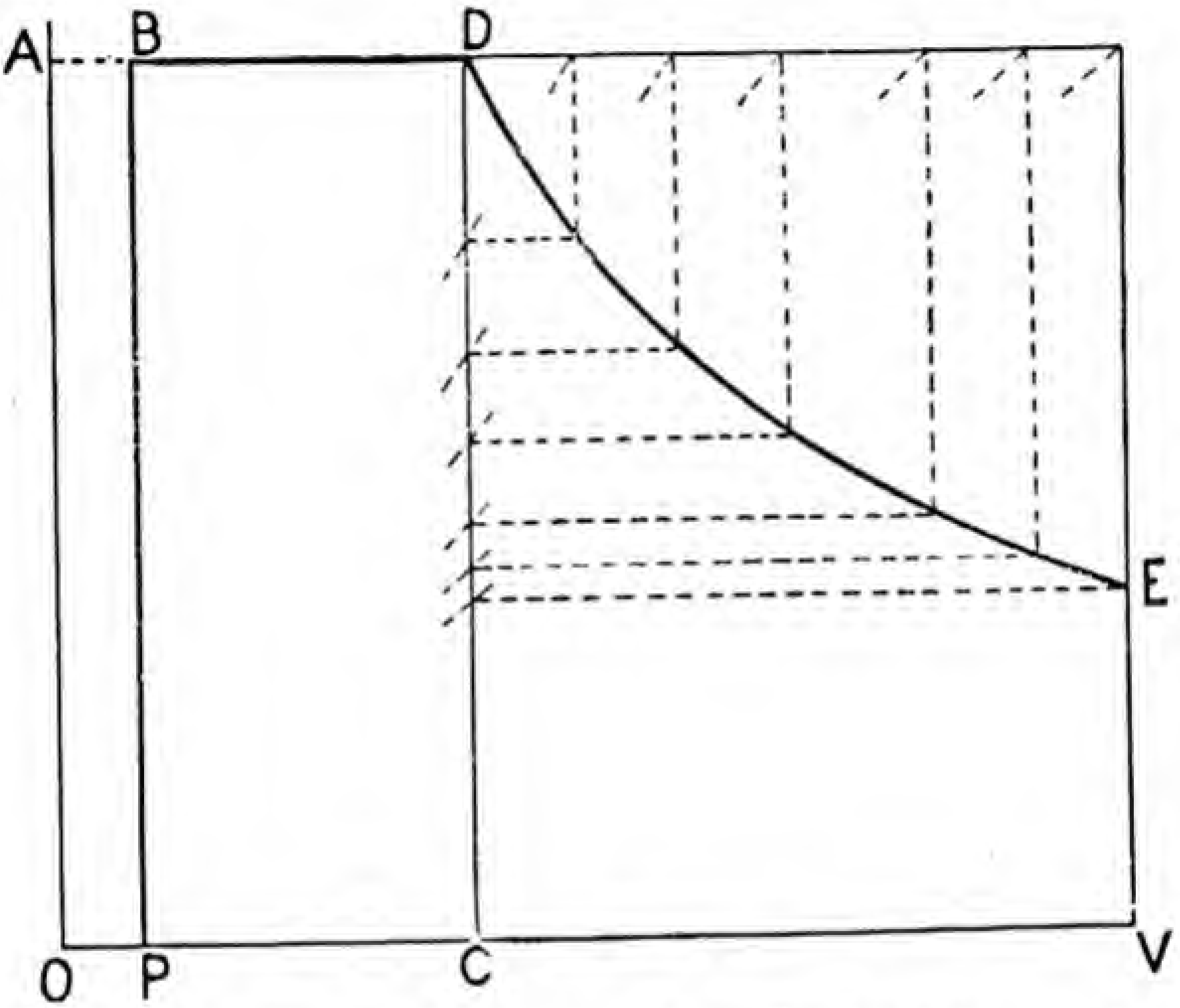


FIG. 63.—Diagram of work, having clearance taken into account.

Take C so that PC is equal to one-third of PV; draw PB and CD perpendicular to PV, making the length of each to represent 60 lbs. per square inch to the scale of force. BD will now represent the admission part of the diagram. Points on the expansion curve may be obtained by calculation as before, or the graphical method explained on p. 49 may be used. This latter method has been used in Fig. 63, giving $PBDEV$ as the required part of the diagram of work. The student will note that the radiating lines required for finding the points on the curve must be drawn so as to converge at O, not at P.

EXERCISES ON CHAPTER VI.

1. Express 1 foot-ton of work in metre-kilograms.

2. How much work in foot-pounds is done per hour by an engine developing 3500 horse-power?

3. A steam engine cylinder has a piston 30″ in diameter × 48″ stroke. Steam is cut off at one half of the stroke, the initial pressure being 105 lbs. per square inch absolute. Assume the simplest law of expansion and draw a diagram to represent the total work done on one side of the piston. Find the mean pressure from your diagram and calculate the work done per stroke. Neglect clearance.

4. Calculate the mean pressure and hence the net work done per stroke for the engine in Exercise 3, using the equation

$$p_m = p_1\left(\frac{1+\log_e r}{r}\right) - p_b.$$

Take the back pressure to be 45 lbs. per square inch.

5. What is meant by the term "hypothetical diagram"? Sketch approximately such a diagram for an engine cutting off at one-third stroke. Indicate on the diagram in what respects the actual engine diagram would probably differ from the hypothetical diagram.

6. An engine having a cylinder 24″ diameter × 36″ stroke has a clearance of 10 per cent. of the volume swept. The point of cut off is 0·6 of the stroke. What are the approximate and true ratios of expansion? Express the clearance volume in cubic inches.

7. Draw the hypothetical diagram for the cylinder of Exercise 6; show the probable diagram by dotted lines.

8. State the following amounts of energy in foot-pounds:

(*a*) A weight of 35 tons may fall vertically 15 feet.

(*b*) The kinetic energy of a projectile of 60 lbs. moving 2000 feet per second.

(*c*) The calorific energy of 1 lb. of coal, 8500 Centigrade pound heat units.

(*d*) 30 lbs. of water raised from 40° F. to 103° F.

(*e*) One horse-power-hour.

(*f*) One kilowatt-hour. 1906.

9. Steam enters a cylinder at 140 lb. pressure (absolute) per sq. inch; is cut off at 0·35 of the stroke and expands according to the law "pv constant." Neglect clearance and cushioning, and draw the hypothetical diagram usually taken. Back pressure 17 lb. per sq. inch. Find the effective pressure. Area of piston, one square foot; stroke, 2 feet. What is the work done in one stroke? How many cubic feet of steam entered the cylinder? What is the work done per cubic foot? 1903.

10. What is meant by "clearance"? If a piston is 12 inches in diameter, and the crank 1 foot, what is the working volume in cubic inches? If the clearance is such that 4 lbs. of water just fill it when the piston is at the end of its stroke, express it as a percentage of the working volume. If the working volume is represented to such a scale that a distance of three inches represents 1 cubic foot, what distance will represent the clearance? 1898.

CHAPTER VII.

THE INDICATOR.

The indicator.—To obtain a knowledge of what is actually occurring in the cylinder of an engine an instrument called an **indicator** is used. The function of this instrument is to draw a diagram on a piece of paper (usually called a **card**) which will show the changes of pressure and volume. There are many different forms of modern instruments. A few well-known types are described here.

The Crosby indicator.—The external appearance of this indicator is shown in Fig. 64. A view of the same instrument is shown partly in section in Fig. 65. The instrument consists of a small cylinder 4 secured to an outer cylinder 5 and fitted with a piston 8 which is held down by a helical spring attached at its upper end to the cylinder cover 2, and at its lower end to the piston. The cylinder 5 is provided with a union 6 by means of which it is attached to an **indicator cock** shown in Fig. 68, which in turn is screwed into a hole made to receive it near the end of the engine cylinder.

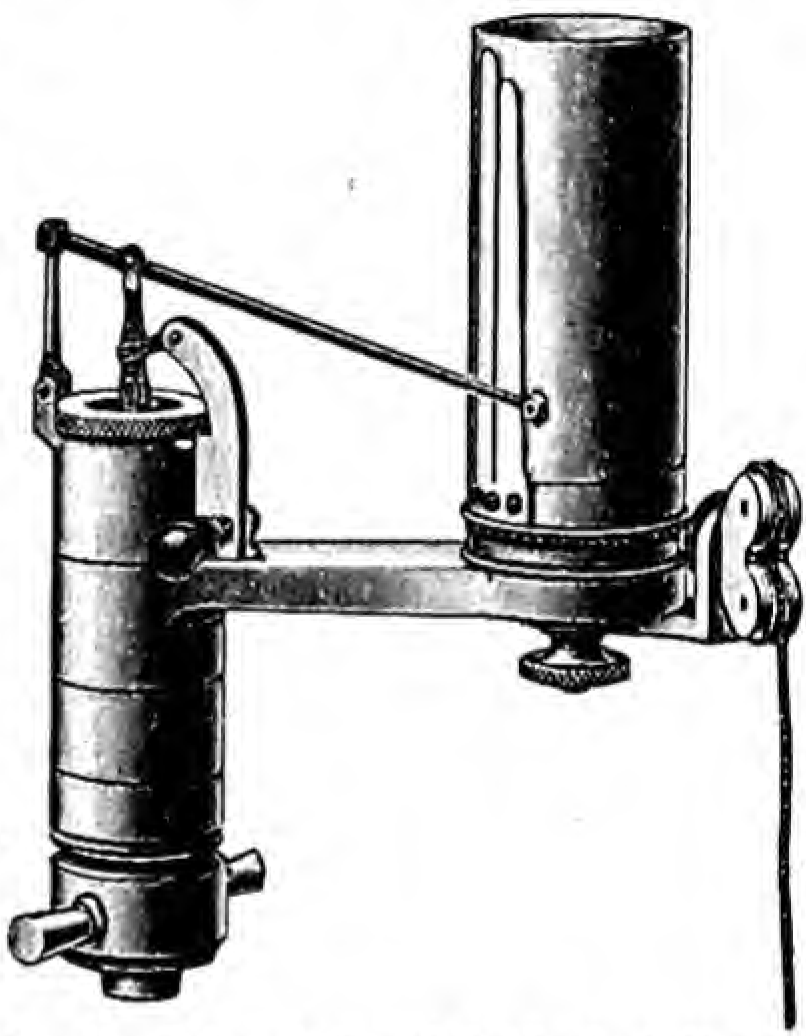


FIG. 64.—External view of the Crosby indicator.

When steam enters the cylinder of the engine, it will also be free to enter the indicator cylinder provided the cock is open, and will exert on the piston 8 the same pressure per square inch as on the piston of the engine. The piston 8 will be forced upward by the steam pressure against the action of the spring. The spring

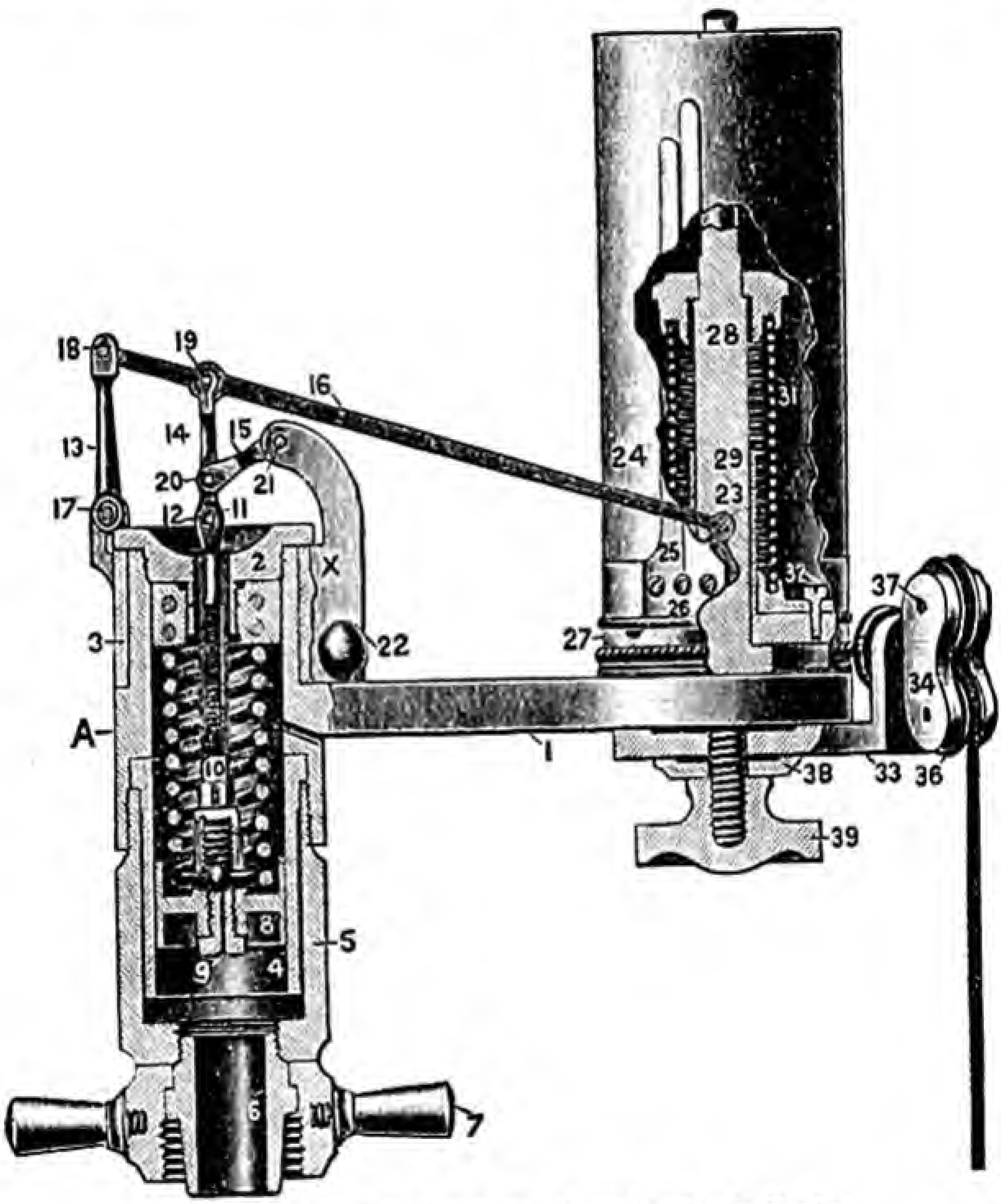


Fig. 65.—Sectional elevation of the Crosby indicator.

follows the well-known law very closely, viz. the amount of compression (or shortening) is proportional to the load applied. It follows, therefore, that the piston 8 will take up a definite position in the cylinder for a given steam pressure. The piston rod is attached at its upper end 12 to a **parallel motion** formed of small bars 11, 13, 14, 15, by means of which the motion of the piston is communicated to a lever 16 in such a manner that its outer end 23 is constrained to move in a vertical line. A pencil is attached to

the lever at 23, and is given, by the arrangement of the mechanism, a faithful copy of the motion of the piston, but on a much larger scale. In this indicator the piston movements are multiplied six times at the pencil. The object is to keep the piston movements as small as possible in order that the pencil movements may be interfered with to a minimum degree by the inertia of the piston.

A drum 24, round which the card is to be wrapped, is mounted so as to rotate on a pin 28 secured to a bracket 1 which is fixed to the cylinder 5. A cord is secured to the lower part of the drum, and, after coiling round the drum is led over guide pulleys 34 to be attached to some suitable part of the engine. A pull on the

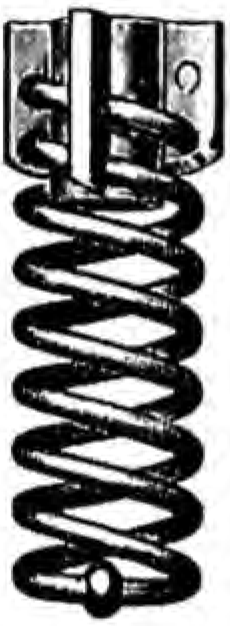


FIG. 66.—Crosby indicator spring.

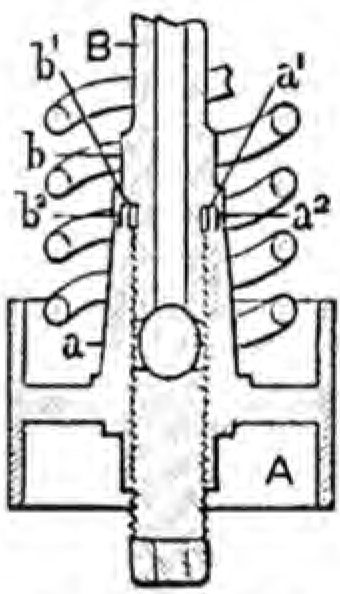


FIG. 67.—Attachment of the spring to the piston in the Crosby indicator.

cord will rotate the drum in one direction, and on releasing the cord, a spring 31 inside the drum is arranged to give the drum rotation in the contrary direction. The cord should be attached to some part of the engine which will give to the paper on the drum a horizontal movement under the pencil 23, exactly corresponding with the movements of the piston in the engine cylinder. It follows, therefore, as the pencil has vertical movements corresponding with the steam pressure in the engine cylinder, that a diagram will be drawn on the paper showing the precise relation of the engine piston position (or volume of steam in the cylinder) and steam pressure throughout the cycle.

The Crosby spring is a double helix (Fig. 66), and has a ball at its lower end fitting a socket in the piston 8 (Fig. 65); the screwed end of the piston rod 10 and another small screw 9 provide the

means for holding the spring and adjusting the socket bearing. The arrangement ensures free and true movement of the piston in the cylinder. The piston 8 is made of tool steel, very thin so as to be as light as possible, and is hardened, ground and lapped to fit. An enlarged section of the piston and arrangement for attaching the spring is shown in Fig. 67. Leakage is prevented past the piston by means of shallow channels cut round its outer surface. These retain moisture and oil, and prevent leakage by setting up eddies in any fluid attempting to flow past, thus offering a greatly increased resistance to the flow. The cylinder cover is formed of a cap 2 screwed to the upper end of the cylinder (Fig. 65). The cap 2 also serves to retain a sleeve 3 in position; this sleeve supports the parallel motion and is free to rotate, the object being to permit of the pencil 23 being brought into contact with or removed from the paper. Drum stops are provided by means of which the movement of the drum is limited to somewhat less than a revolution.

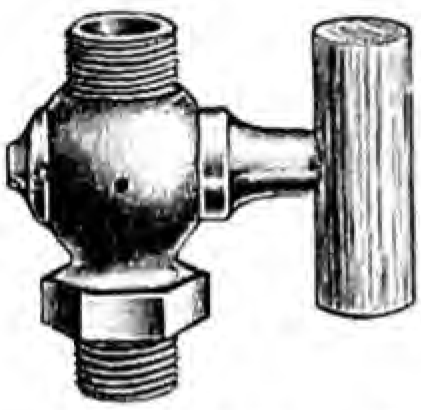


FIG. 68.—Indicator cock.

The indicator cock (Fig. 68) is open when the handle is placed straight. On placing the handle crosswise, the indicator cylinder is shut off from the engine cylinder and is put into communication with the atmosphere, through the small hole shown in the centre of the ball. The pressure on the indicator piston will then be that of the atmosphere. A line drawn by the pencil on the card under these conditions will show atmospheric pressure and will serve as a datum line for measuring other pressures on the diagram.

Thompson indicator.—This indicator is illustrated partly in section in Fig. 69. Messrs. Schaffer & Budenberg, Ltd., are the makers. The student will have no difficulty in understanding the arrangement of the principal parts after what has been said regarding the Crosby indicator. In the Thompson indicator, as in the Crosby, the piston is of steel, the springs are double-wound and are ball-jointed to the piston and rod, and the cylinder is fitted with an inside liner which forms a steam jacket between the liner and the outer cylinder. The movement of the piston is multiplied six times at the pencil. The arrangement of the

parallel motion differs from that of the Crosby indicator as will be seen by an inspection of the illustrations.

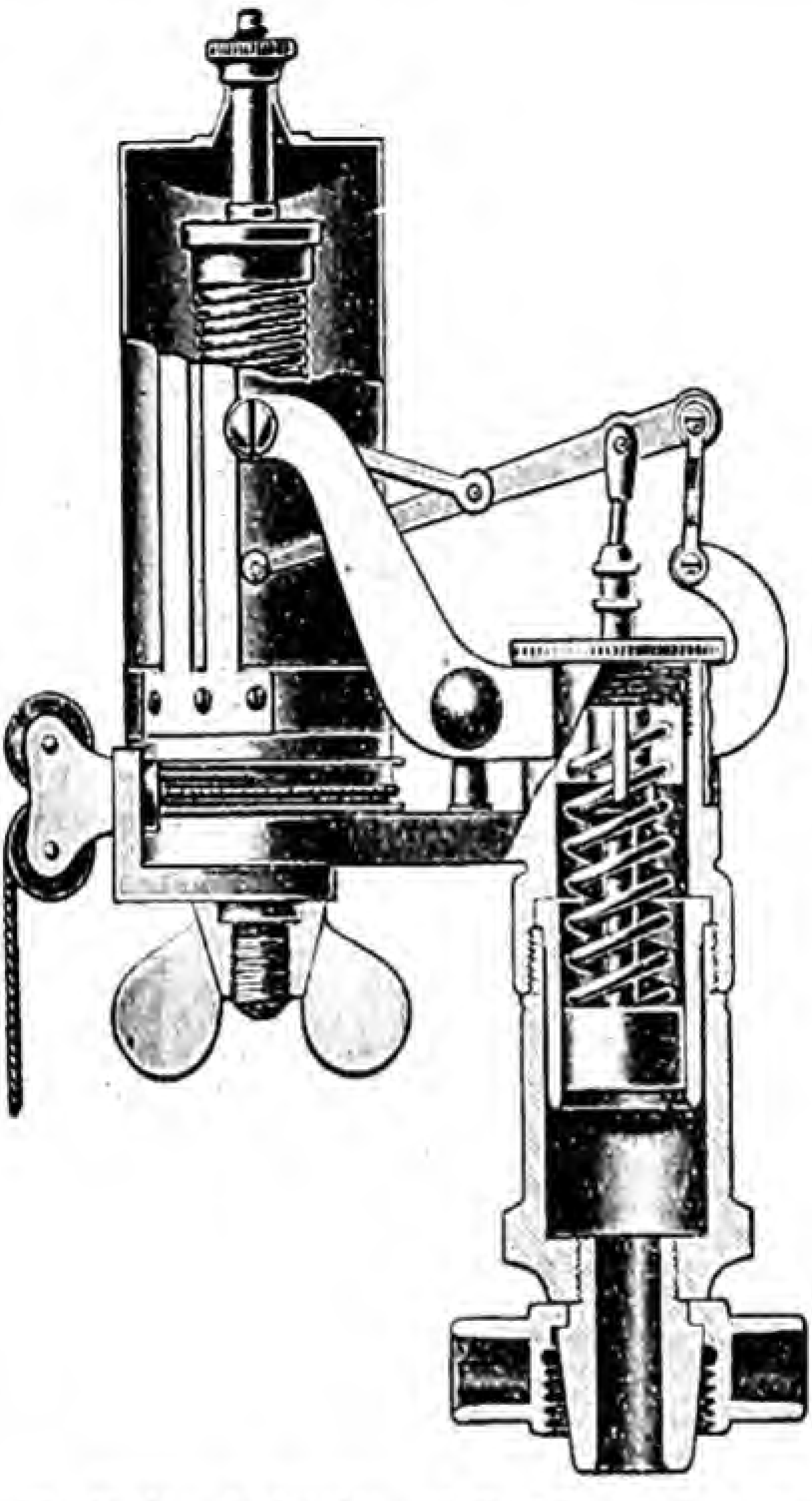


FIG. 69.—Sectional elevation of the Thompson indicator.

The Simplex indicator.—This indicator differs materially from those previously described. The makers are Messrs. Elliott Bros. The spring is tongs-shaped, one limb of the tongs being attached to the top of the piston rod (Fig. 70), and the other limb to a fixed bracket. The spring is outside the indicator cylinder when in position and is thus removed from the heating action of the steam. This is an important feature, as the strength of a spring is not constant at all temperatures, and, if the temperature of indicator springs be allowed to vary, errors will be introduced which will affect the pressures shown on the resulting diagram. The spring in the Simplex indicator is easily changed without

removing any other part, and being cool is easily handled. The piston, piston rod, and parallel motion are so arranged that they can be removed for cleaning in one piece, by unscrewing the milled cover shown just under the tongs-spring in Fig. 70.

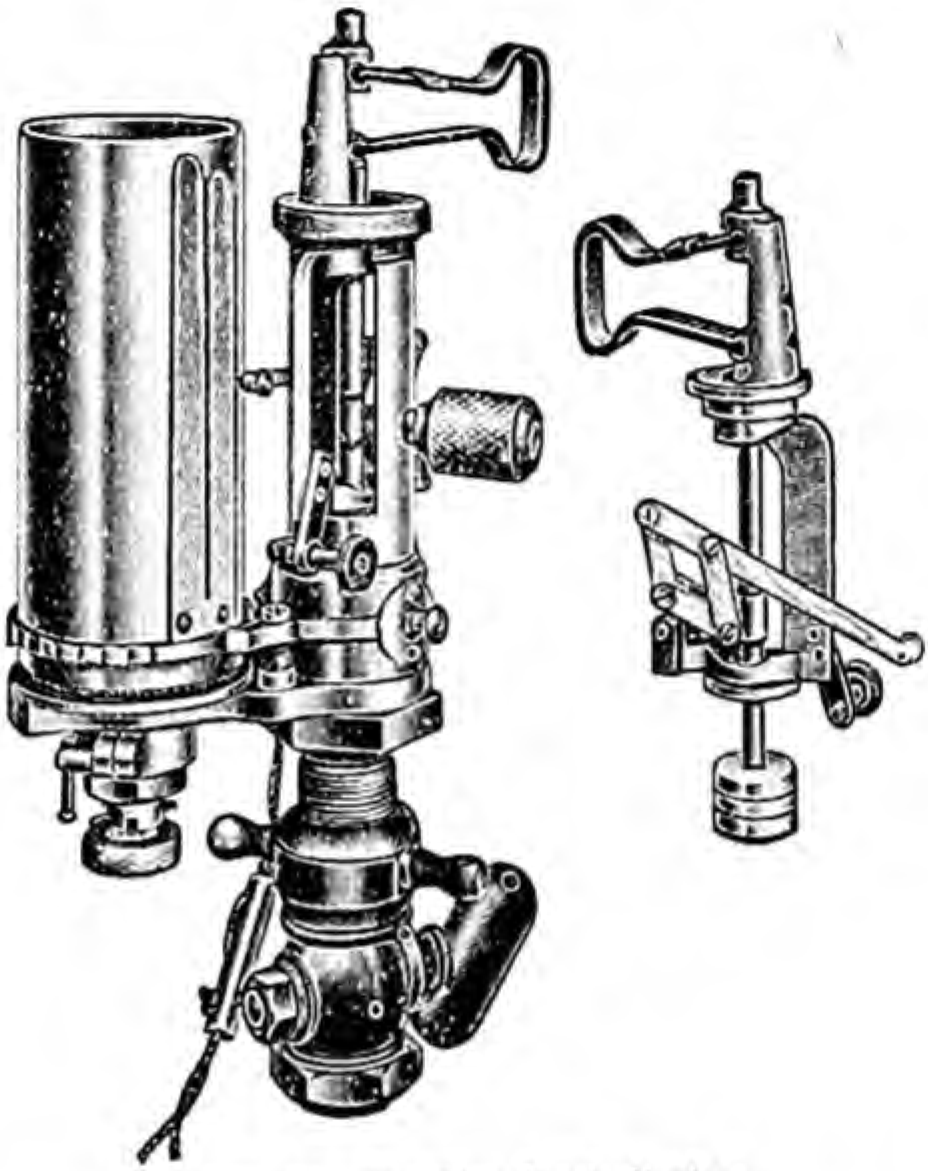


FIG. 70.—The Simplex indicator.

The "McInnes-Dobbie" indicator.—This indicator is illustrated in Fig. 71. The instrument is an excellent example of the type in which the spring for controlling the piston is helical and placed outside the cylinder. Referring to the section in Fig. 71, *A* is the helical drum spring, having its end a^1 fixed to the spindle square *B*, while its other end a^2 reciprocates with the drum. The drum may be removed by unscrewing the milled head at the top of the drum spindle *C*, giving access for adjusting the strength of the drum spring. This adjustment is accomplished by lifting a^1 off the square, when the strength of the spring can be increased or diminished by winding a^1. It will be understood that engines running at high speed of revolution require a greater strength of drum spring than others running at low speed. This is necessary in order that the drum spring may return the drum promptly

without allowing any slack to take place in the driving cord. *D* is the cord pulley frame, made so that the pulley may be adjusted to

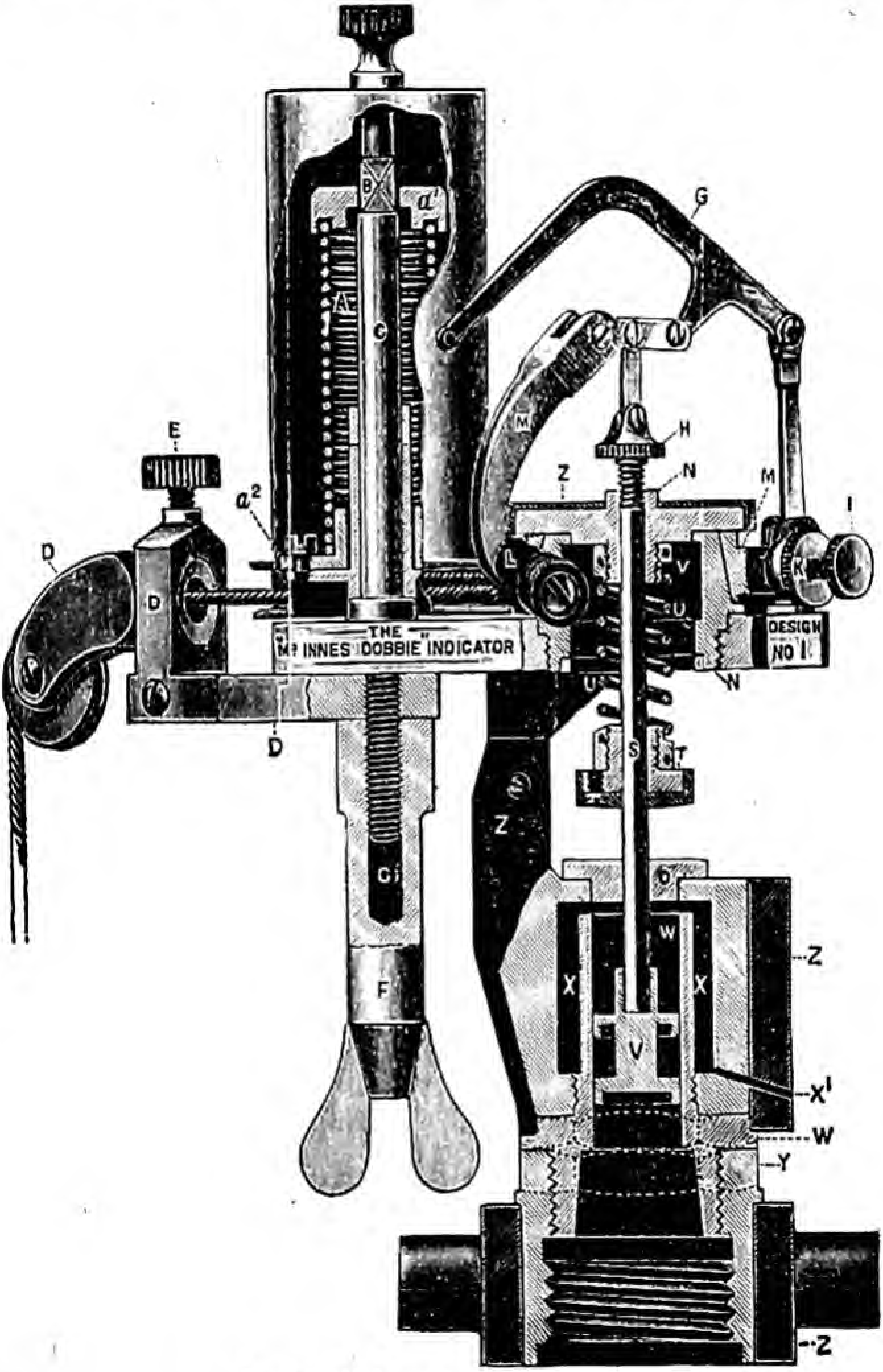


FIG. 71.—McInnes-Dobbie indicator.

any angle for the purpose of carrying the cord to the required part of the engine.

The steel piston is shown at V, the cylinder being marked X. The cylinder is connected to the indicator cock (not shown in Fig. 71) by the coupling nut at the bottom. The piston has a recess providing accommodation for lubricant and acting as a means for catching grit. Any steam leaking past the piston escapes through the passage X^1. The cylinder is sheathed with vulcanite Z, thus enabling the indicator to be handled without burning the fingers.

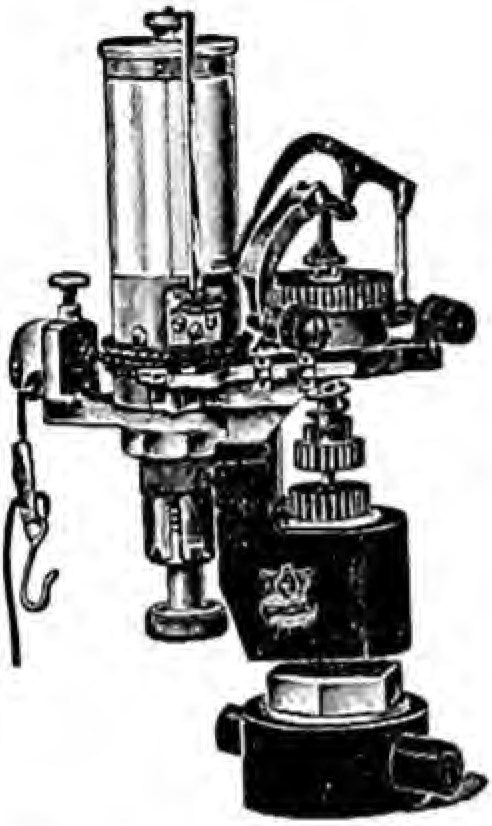


FIG. 72.—McInnes-Dobbie indicator, having paper roll inside the drum.

FIG. 73.—McInnes-Dobbie indicator, arranged for taking continuous diagrams automatically.

The pencil lever is marked G and is connected by the parallel motion levers to a milled nut H on the top of the piston rod S; the piston rod is made of steel tube for lightness. The spring U has one end screwed to the top cover N and the lower end screwed to the spring seat T, which is fixed to the piston rod. The bracket M carries the parallel motion, and can swing round the top cover N for the purpose of bringing the pencil to the paper. The movement of the piston is multiplied six times at the pencil.

In Figs. 72 and 73 external views are given of two other patterns of indicators made by the same firm. The drum in each example is made to contain a roll of diagram paper. In the indicator shown in Fig. 73, the paper is fed automatically at each reciprocation of

the drum, thus enabling a continuous sequence of diagrams to be taken (Fig. 74). Useful information may thus be obtained regarding the performance of engines in which the load varies greatly.

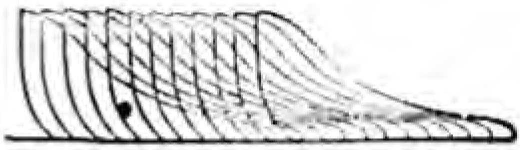


FIG. 74.—Example of continuous indicator diagrams.

Connecting up the indicator.—The indicator is usually connected to the cylinder by means of a bent pipe *E* (Fig. 75), which is connected to both ends of the engine cylinder and has a branch near the middle of the pipe for mounting the indicator. A three-way cock *F*, at the branch connection, enables either end of the cylinder to be connected to the indicator so that diagrams may be

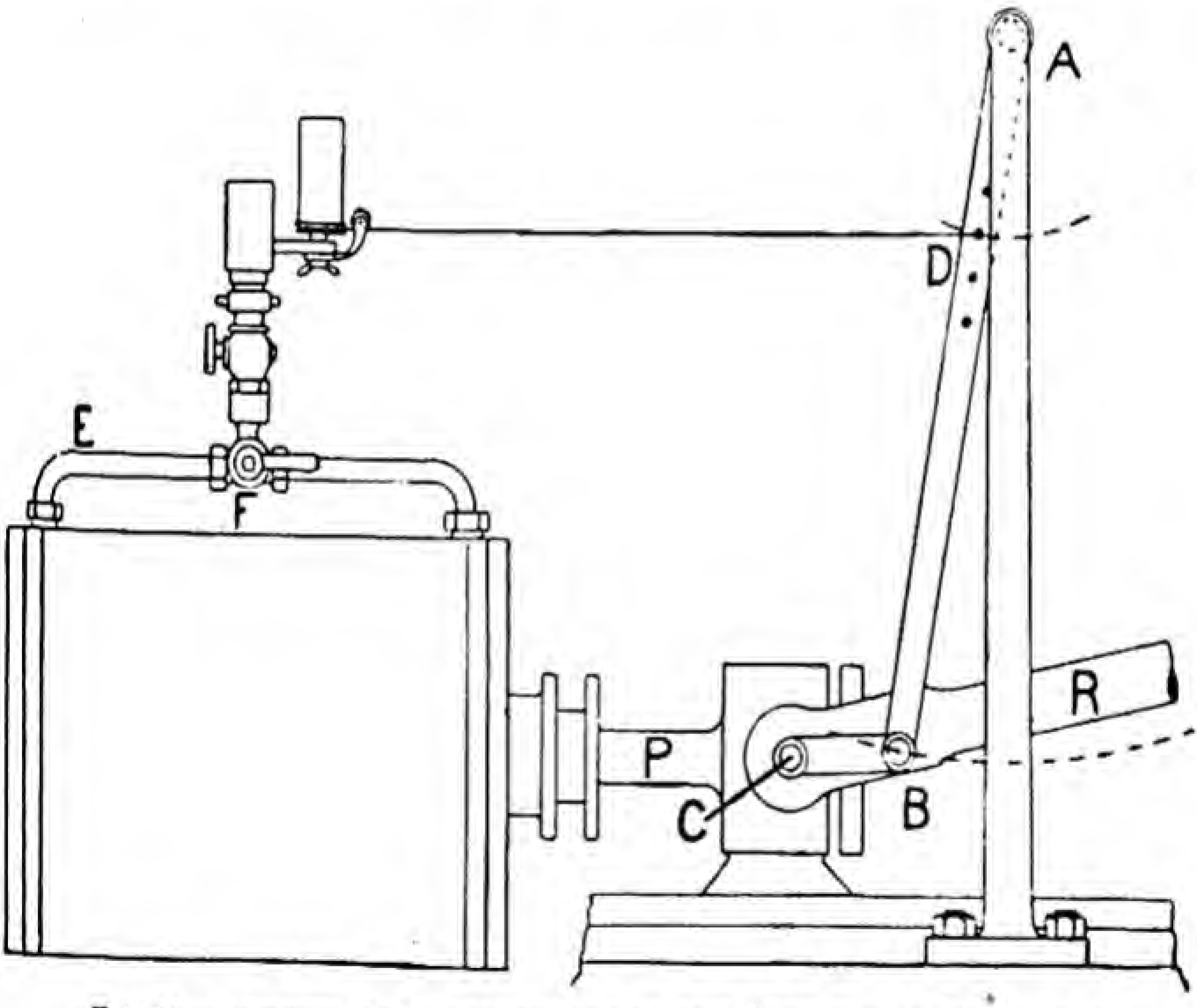


FIG. 75.—Arrangement for connecting up and driving an indicator on a steam engine.

taken from either end of the cylinder. Frequently in testing engines for scientific purposes, two indicators are employed, one connected direct to each end of the cylinder. This plan has the advantage that losses of pressure due to long connecting pipes and bends in them are greatly reduced. The connecting pipes are best placed so as to enter the cylinder at right angles to the steam

ports, thus preventing false pressures, due to the rush of steam into and out of the cylinder, being shown.

Usually the stroke of the engine is too great to permit of the drum being driven by direct connection to the crosshead. A **reducing gear** has generally to be employed. A simple form of such gear is shown in Fig. 75. A rod AB has one end A pivoted to a fixed axis and its other end B connected to the centre of the crosshead pin by means of a short link BC. The cord driving the indicator drum is connected to the link AB by fastening it through one or other of the small holes at D. In Fig. 75, P is the piston rod and R the connecting rod.

The function of the reducing gear is to give to the drum a faithful copy of the motion of the piston on a reduced scale. Too many joints in such gears are objectionable on account of their liability to work loose.

The indicator in use.—In using an indicator, attention should be given to the following points:

1. See that the indicator is mounted properly, and that the joints of the connection to the cylinder pipes do not leak. Arrange the position of the drum and guide pulleys so that the cord runs freely when connected to the reducing gear.

2. Adjust the length of the cord so that the drum does not knock against the stops at either end of its stroke.

3. Choose a spring of proper strength to suit the maximum pressure to be expected.

4. See that the pencil is sharp; but in the case of a metallic pencil, the paper may be cut if the pencil is too sharp.

5. Oil the piston before inserting it. If in use during a long test, remove the piston occasionally for cleansing and oiling.

6. When finished, remove the indicator immediately and clean it thoroughly before putting it away. Do not leave the spring in the cylinder; it should be cleansed and oiled to prevent corrosion.

Taking diagrams.—The following procedure may be followed:

1. Fold over about $\frac{1}{4}''$ of one short edge of the paper. Insert this edge in one of the drum clips. Bend the paper round the drum and insert the opposite edge in the other clip. Pull the paper down the drum and see that it is pressed tightly against it. Fold the short edges over the clips.

2. Connect the drum cord to the reducing gear.

3. Bring the pencil to the paper and see that the pencil is so adjusted that undue pressure does not occur, otherwise the friction of the pencil will be excessive and will introduce errors in the pressures shown.

4. Connect the indicator cylinder to the atmosphere by means of the cock and draw the atmospheric line.

5. Open the cock communication to one end of the cylinder and allow the indicator piston to rise and fall several times so as to warm the indicator cylinder. Turn the indicator cock two or three times open to the atmosphere so as to blow any water out.

6. Bring the pencil to the paper to draw the diagram. Repeat the operations for the other side of the engine cylinder.

7. Disconnect the drum cord and remove the paper. Be careful that dirty fingers, steam, oil and water do not touch the diagram on the paper.

8. Write at once the following information on the card:
   - (*a*) Date and time card was taken.
   - (*b*) Engine from which card was taken.
   - (*c*) Scale of spring.
   - (*d*) Revolutions per minute at time card was taken.
   - (*e*) Diameter and stroke of piston.

Indicated horse-power.—Fig. 76 shows a pair of indicator diagrams taken from the two sides of the piston of a steam engine. From these the mean pressures exerted on the opposite sides of the piston may be determined by the method explained on p. 80. Having ascertained the mean pressures, the work done on the piston during each forward and backward stroke can be calculated; and, knowing the number of such double strokes per minute, the work done on the piston per minute can be estimated. If this result, stated in foot-pounds, be divided by 33,000, the calculation will give the horse-power developed on the piston of the engine. This is called the **Indicated Horse-Power** (written generally I.H.P.), as it depends on a knowledge of the mean pressure obtained by use of the indicator.

Calculation of I.H.P.—The mean pressures have been deduced from the diagrams in Fig. 76 by first drawing lines touching the extremities of the diagrams and perpendicular to the atmospheric line *AL*. The distance between these is then divided into

ten equal parts and an ordinate drawn from the middle of each part. By this process the same points of division serve for both diagrams. The heights of these ordinates measured between the curves of each diagram are written down, using the scale of the spring, those belonging to the left-hand diagram being placed above the curves, the others being below. The sums of these columns divided by 10 give the mean pressure for each side of the piston.

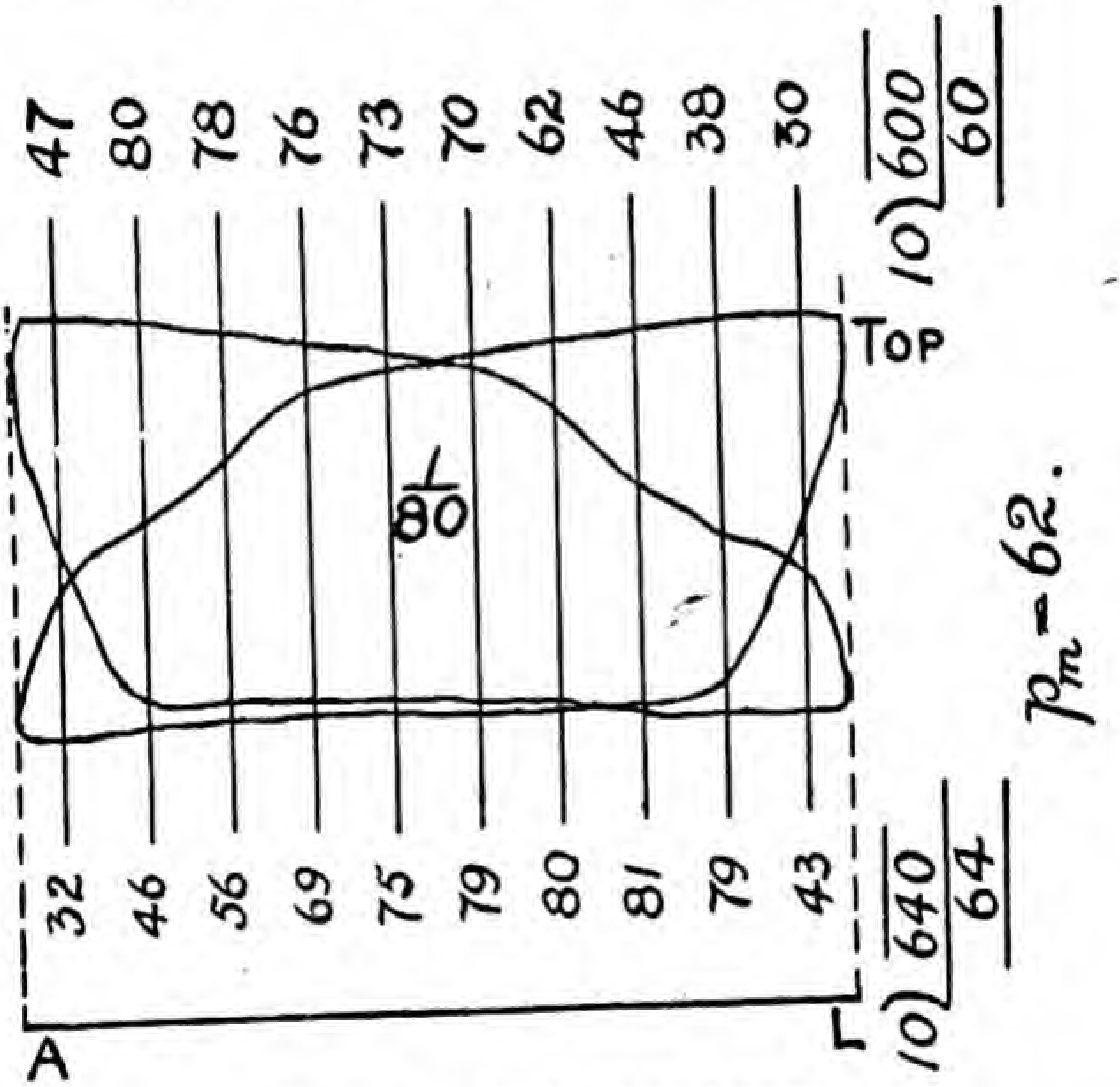


FIG. 76.—Example of an indicator card.

As a **first approximation**, the average value of the mean pressures so obtained may be calculated by taking their sum and dividing by 2. Using this as an average pressure common to both sides of the piston and writing it p_m lbs. per square inch, we have:

Work done on piston per stroke $= (p_m \times A \times L)$ ft.-lbs.

where A = area of piston in square inches,

L = length of stroke in feet.

Let N = revolutions per minute,

then $2N$ = number of strokes per minute;

$\therefore$ work done on piston per minute $=(p_m A L 2 N)$ ft.-lbs.

$$\text{I.H.P.}=\frac{2p_m ALN}{33{,}000}. \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

The result so calculated is approximate only, as no allowance has been made for the effect of the piston rod in reducing the area of that side of the piston to which it is attached. Thus, by reference to Fig. 77, it will be seen that the whole area of the piston on side A, viz. $\frac{\pi D^2}{4}$ is acted on by the steam pressure, while on side B the net area available to be acted on by the steam is the difference between the area of the piston and the area of the piston rod, viz. $\left(\frac{\pi D^2}{4}-\frac{\pi d^2}{4}\right)$, D being the diameter of the cylinder, and d that of the piston rod, both stated in inches.

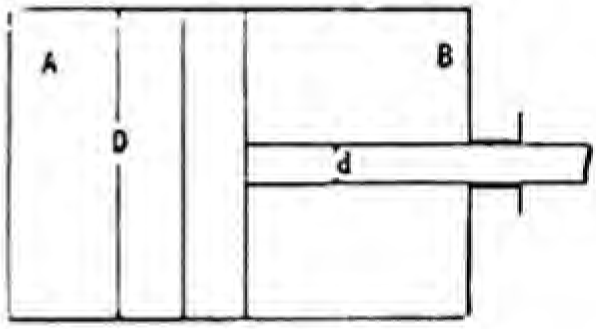


FIG. 77.

To obtain a more accurate result we may proceed thus:

Let p_A = average pressure for end A of cylinder lbs. per square inch,
p_B = " " " B " " "

$$\text{Total pressure on side } A=p_A\times\frac{\pi D^2}{4}\text{ lbs.,}$$

$$\text{Work per stroke for side } A=p_A\times\frac{\pi D^2}{4}\times L\text{ ft.-lbs.,}$$

$$\text{I.H.P. for side } A=\frac{p_A\times\frac{\pi D^2}{4}\times L\times N}{33{,}000}. \quad\ldots\ldots\ldots(2)$$

N being the revolutions per minute, and consequently the number of strokes for side A per minute. Again,

$$\text{Total pressure on side } B=p_B\left(\frac{\pi D^2}{4}-\frac{\pi d^2}{4}\right)\text{lbs.,}$$

$$\text{Work per stroke for side } B=p_B\left(\frac{\pi D^2}{4}-\frac{\pi d^2}{4}\right)L\text{ ft.-lbs.,}$$

$$\text{I.H.P. for side } B=\frac{p_B\left(\frac{\pi D^2}{4}-\frac{\pi d^2}{4}\right)L\,.\,N}{33{,}000}. \quad\ldots\ldots\ldots(3)$$

The total indicated horse-power will be obtained by taking the sum of these results, (2) and (3).

Other uses of the indicator.—The indicator is also used to ascertain whether the distribution of steam in the cylinder is as intended; from the information thus derived the valve gear can be altered so as to correct too early or too late admission, cut off, release, or cushioning. The presence of excessive throttling or wire-drawing can also be detected. Such information as can be thus deduced from the diagrams is particularly useful in the case

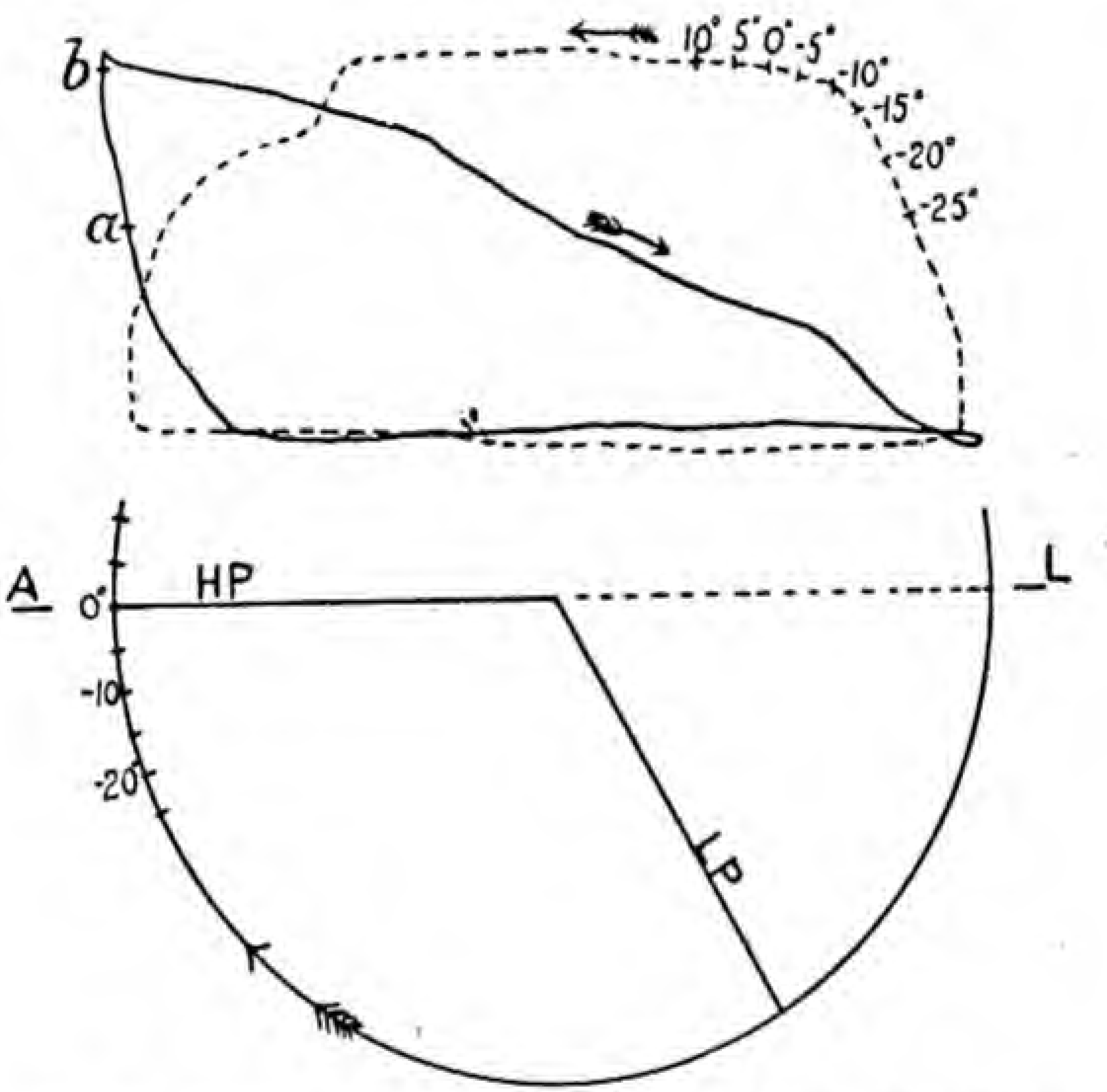


FIG. 78.—Cross diagram for spreading out the "lead" portion.

of testing internal combustion engines (Chap. XXI.). The indicator can also be connected to the steam chest, the drum being driven from the crosshead as usual, giving useful information as to the fluctuation of pressure of the steam in the chest.

Cross diagrams.—It will be observed that the ordinary indicator diagram does not show clearly the state of the "lead" portions of the cycle. Thus, in Fig. 78 is shown an indicator diagram in full lines taken by Mr. A. T. Quelch from the H.P. cylinder of the triple expansion engines of S.S. *Anglo-Saxon.* The part of the diagram showing the cylinder pressures while the H.P. crank moves from $-25°$ through the top dead point to 10° on the other side of

the dead point is included between ab, and the "lead" is quite obscured.

To show the lead clearly, another diagram (shown dotted) was taken in which the indicator was still connected to the H.P. cylinder, but the drum was driven from the L.P. crosshead. The L.P. crank being 120° behind the H.P. crank, the resulting diagram has the lead portion spread out considerably, as shown in Fig. 78, the crank positions and points on the dotted indicator card being numbered to correspond. The artifice is often adopted in gas and oil engine work for spreading out the explosion part of the diagram. In such cases the drum is driven from a small crank placed at 90° to the engine crank.

Brake horse-power.—A considerable portion of the horse-power developed on the piston is not available for driving outside machinery. Generally from 5 to 25 per cent. of the I.H.P. is used in overcoming frictional resistances in the mechanism of the engine itself. To obtain an estimate of the horse-power which the engine can deliver, the energy delivered by the engine may be absorbed by means of a brake mounted on the fly-wheel. A common form of brake is illustrated in Fig. 79, and consists of a double rope passed round the wheel and held in position by means of loosely fitting wood blocks secured to the rope. Four such blocks are shown in the illustration, a separate view being given of one to indicate the manner in which they embrace the rim of the wheel. One end of the rope is attached to a spring balance suspended from overhead, the other end carries a load W lbs. The direction of rotation of the wheel being as shown by the arrow, it will be noticed that the pull, P lbs., of the spring balance is helping to turn the wheel and W is opposing its rotation. The friction between the ropes and the rim communicates these forces to the wheel.

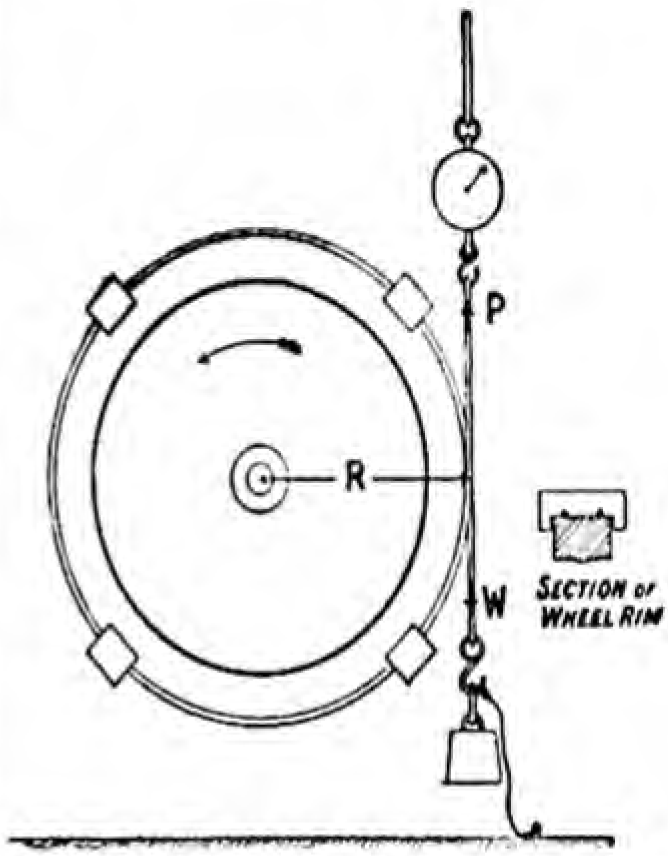


FIG. 79.—A common form of rope brake.

Let R=radius, in feet, to the centre of the rope, then

Net resistance against which the wheel is working $= W - P$ lbs.

As this resistance is applied at a radius R, the distance through which it is overcome in one revolution will be

$$2\pi R \text{ feet.}$$

$$\therefore \text{ Work per revolution} = (W - P)2\pi R \text{ ft.-lbs.}$$

For N revolutions per minute,

$$\text{Work per minute} = (W - P)2\pi R \,.\, N \text{ ft.-lbs.,}$$

$$\text{Horse-power} = \frac{(W - P)2\pi RN}{33{,}000}.$$

The result of this calculation, called the **Brake Horse-Power** of the engine (written B.H.P.), gives the horse-power which the engine can deliver for driving machines, etc.

To prepare the student for the Engine Trials described in Chaps. XVIII. and XXI., the following preliminary experiments should be carried out. It is assumed that a steam or other engine is available.

Expt. 19.—Practise attaching the indicator and its gear several times with the engine at rest. Rotate the crank shaft by hand in order to test the adjustments.

Expt. 20.—Practise mounting the brake with the engine at rest.

Expt. 21.—Take several indicator diagrams. Note also the other particulars required. Work out each diagram for I.H.P.

Expt. 22.—While other students are performing Expt. 21, practise reading the brake loads and other necessary measurements. Work out the B.H.P.

Mechanical efficiency.—The efficiency of any machine is defined as the ratio of the energy which may be obtained from the machine to the energy supplied to the machine. Applying this definition to the mechanism of an engine, the energy supplied to the piston per minute will be obtained by multiplying the I.H.P. by 33,000: also the energy which may be obtained from the engine in one minute will be the product B.H.P. × 33,000. The efficiency of the mechanism, or **mechanical efficiency** as it is called will be

$$\text{Mechanical efficiency} = \frac{\text{B.H.P.} \times 33{,}000}{\text{I.H.P.} \times 33{,}000}$$

$$= \frac{\text{B.H.P.}}{\text{I.H.P.}}.$$

The result is generally multiplied by 100 so as to have it as a percentage. The mechanical efficiency generally lies between 75 and 95 per cent.

Heating of the brake.—In **absorption dynamometers** such as that illustrated in Fig. 79, the energy produced by the engine is absorbed by the frictional resistances of the brake, and is transformed into heat. It is therefore necessary to keep the fly-wheel cool by lubrication with soapy water, this cooling being assisted by the air draught produced by the rotation of the wheel. Sometimes fly-wheels have their rims made of a channel section so as to receive a stream of water, which being whirled round by the wheel, retains its position in the rim in the same way as a whirled stone at the end of a string keeps its circular path. The water is kept continually flowing into the rim and is drained away by a sharp-edged scoop on the other side, and therefore keeps the rim cool. The arrangement is shown in Fig. 80.

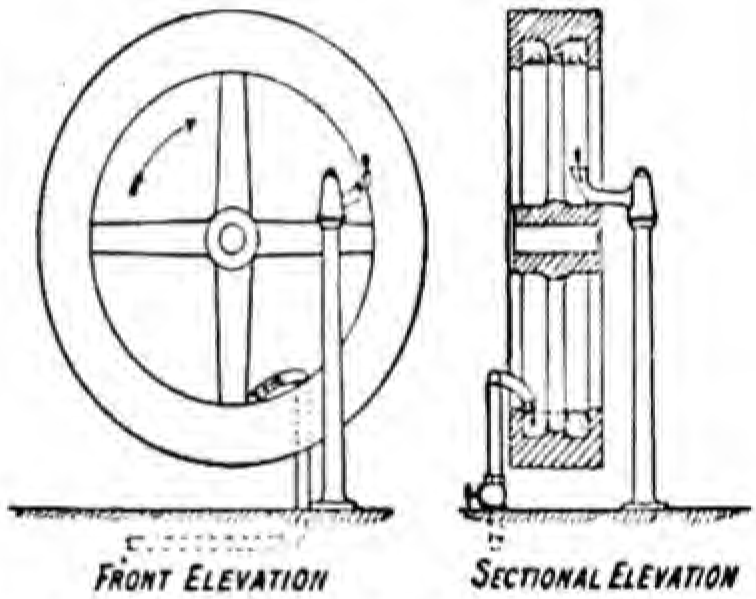


FIG. 80.—Arrangement for water-cooling the rim of a brake wheel.

Measurement of large powers.—It is not practicable to apply a brake to any engine of large power. The power available for driving purposes in such cases may be estimated from a knowledge of the I.H.P. obtained from indicator diagrams, using an estimated value for the mechanical efficiency. This procedure is often followed in marine engines. If the engine is used for driving electric generators it is possible to estimate very closely the power coming from the engine by calculating the output from the dynamo and then allowing for the losses in the dynamo. Thus,

Power coming from the engine = output from dynamo + losses in the dynamo.

The **electrical horse-power** (E.H.P.) delivered by the dynamo may be calculated from

$$\text{E.H.P.} = \frac{\text{amperes} \times \text{volts}}{746}.$$

The electrical and mechanical losses in the dynamo may be experimentally determined, the methods of experimenting being, however, beyond the scope of this book.

The electrical method is particularly valuable in the case of steam turbines. Such motors have no reciprocating piston, but consist of wheels having many blades. The steam flows over these blades and so rotates the wheels (see Chap. XVII.). It is impossible to derive any information regarding horse-power from turbines by use of an indicator, hence the value of the information to be obtained from the electrical horse-power of dynamos driven by steam turbines.

Many vessels are now fitted with steam turbines for driving their propellers, and two are now (1907) being built for the Cunard Co., in each of which the total horse-power of the steam turbines will amount to 75,000. Recently an ingenious method of determining the horse-power of marine turbines has been applied, consisting of measuring the angle through which one end of a length of the propeller shaft twists relative to the other end when the turning moment is applied by the engine. The angle of twist is proportional to the turning moment, and, when it is known, the turning moment may be calculated from it and a knowledge of the elastic constants of the material of the shaft. The revolutions having been ascertained, the horse-power coming from the engines may be determined.[1]

EXERCISES ON CHAPTER VII.

1. Give an outline sketch of the parallel motion, piston, and piston rod of any indicator you have handled.

2. Sketch in section and describe in detail the cylinder and piston of any indicator.

3. Give sketches and description of any kind of reducing gear for connecting up an indicator to the engine crosshead.

4. In a steam engine cylinder, the mean pressure was found to be 26·7 lbs. per square inch for one side of the piston and 28·2 lbs. per square inch for the other side. The piston is 24″ diameter × 36″ stroke.

---

[1] See articles "Torsiometers," by Mr. Archibald Denny, and "Torque of Propeller Shafting," by Mr. J. Hamilton Gibson; *Engineering*, April 12th, 1907.

Calculate the I.H.P. when running at 95 revolutions per minute. Neglect the effect of the piston rod.

5. Answer question 4 again, taking account of the piston rod, which is 4″ in diameter and is on that side of the piston which has 28·2 lbs. per square inch acting on it.

6. Give a sketch and description of any form of brake which you have seen used for obtaining the B.H.P. of an engine.

7. In testing an engine for B.H.P. a brake similar to that shown in Fig. 79 was used. The speed of the engine was 230 revolutions per minute; the pull of the spring balance 15 lbs., and the dead load 84 lbs. The brake wheel was 4′ 9″ diameter. Calculate the B.H.P.

8. In question 7, calculate the heat developed by the brake. State the result in B.T.U. per minute, taking $J=778$.

9. In question 7 the mechanical efficiency was found to be 82·5 per cent. What was the I.H.P.? How much energy in foot-lbs. per minute is used in overcoming frictional resistances of the engine?

10. Describe an indicator; how is it attached to a steam, or gas, or oil engine? Choose some one of these and sketch the sort of diagram obtained, and state what information it gives us. Show how the horse-power is calculated. 1898.

11. Describe, with sketches, how you would take an indicator diagram of a steam engine or a gas engine. Sketch a possible diagram and explain how you would calculate the indicated horse-power. What information is necessary? 1905.

12. The mean effective pressure on the piston, both in the forward and back strokes, is 62 lbs. per square inch; cylinder 18″ diameter; crank 18″ long. What is the work done in one revolution? 1906.

CHAPTER VIII.

VALVES AND VALVE GEARS.

Position of the piston.—To obtain the position of the piston in the cylinder, the following construction may be followed. Describe a circle with centre C and radius CB, representing the length of the crank to a convenient scale (Fig. 81). Take RL produced as

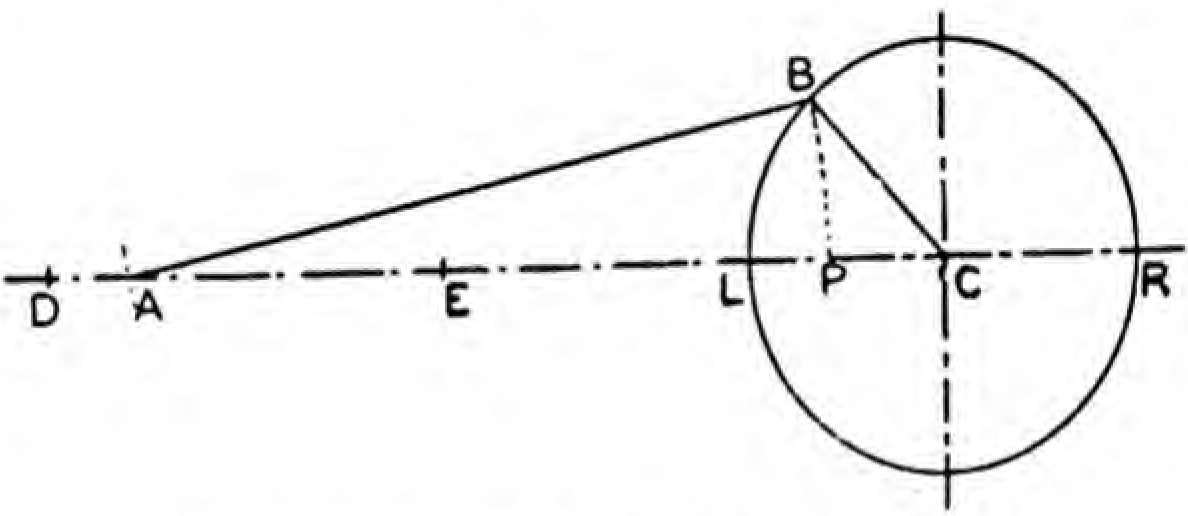


FIG. 81.—Construction for finding the position of the piston.

the centre line of the cylinder. Draw the crank in any position, such as CB. With centre B and radius BA, equal to the length of the connecting rod to scale, cut the centre line in A. A will now be the position of the centre of the crosshead pin, corresponding to the crank position CB. A will travel from D to E and back again as the crank rotates; its distance from the beginning of the stroke in the given position is DA, and the piston will, of course, be at a distance equal to DA from the commencement of its stroke. The crank is said to be on the **dead points** when it is passing through the positions CL and CR. L is called the **inner** dead point, being nearer to the cylinder than R, which is called the **outer** dead point.

With centre A and radius AB, describe an arc cutting LR in P. Then LP will be equal to DA, and consequently will show the distance travelled by the piston from the beginning of the stroke. Draw BM perpendicular to LR (Fig. 82). Notice, in carrying out the above construction, that if the connecting rod is made longer, the point P will lie closer to M. In the particular case of the connecting rod being made infinitely long, P will coincide with M.

With the comparatively short connecting rods used in ordinary engines, the piston position must be found as at P by the method described, but in the case of connecting rods which are very long as compared with the crank radius, the position may be assumed to be M without serious error.

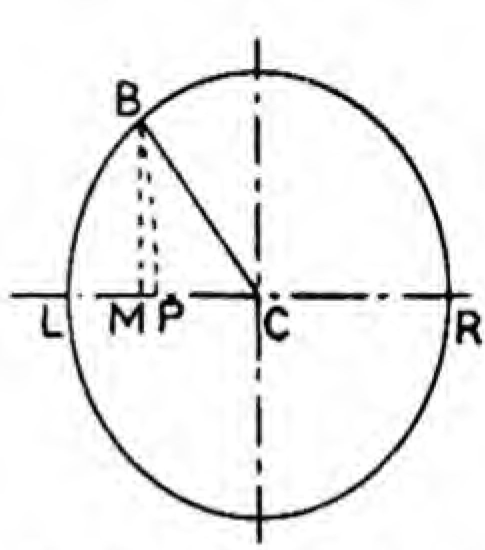


Fig. 82. – Construction for finding the position of the slide valve.

Position of the slide valve.—Usually the slide valve is driven by an eccentric. Imagine an ordinary crank of small radius to have the crank pin made very large, of such diameter, indeed, that the crank cheek disappears and the hole for receiving the crank shaft may be bored through the crank pin. We have now an eccentric, and the motion which it will give to the valve will be precisely the same as that which the original crank would have communicated. Valve positions may therefore be found by the same method as that described above for piston positions.

Usually the eccentric rod connecting the eccentric to the valve spindle is long when compared with the eccentric radius. It may, therefore, be assumed in most cases that the valve position is that shown by the point M in Fig. 82.

Events determined by the valve.—For each side of the piston the valve determines the instant at which four events will occur. These are:

(1) **Point of admission.**—This is the instant at which steam begins to flow into the cylinder, and occurs when the crank is still a few degrees from reaching the dead point. The valve is thus open a little before the piston is ready to begin its stroke, and thus a supply of steam has had time to enter the cylinder and

fill the clearance space between the piston and the steam port; this is said to be giving **lead** to the valve.

(2) **Point of cut-off.**—This point may occur when the piston has completed any convenient fraction of its stroke. At the required instant, the valve closes the steam port, and the remainder of the stroke is completed by the expansive action of the steam.

(3) **Point of release.**—This is the point at which exhaust from the cylinder begins. To permit a rapid discharge of the steam when done with, and so to reduce the back pressure on the piston, it is customary to open to exhaust some little distance before the piston reaches the end of its stroke.

(4) **Point of cushioning.**—At this point the exhaust port closes and no further steam is allowed to flow from the cylinder. Usually about 0·8 of the return stroke has been completed when this occurs, and the steam entrapped in the cylinder is compressed by the returning piston and so has its pressure raised. The object of this arrangement is to provide a cushion of steam to bring the piston to rest at the end of the stroke and so to prevent the bearings knocking; cushioning also fills the clearance spaces with steam at pressure higher than that of the exhaust steam, and so reduces the quantity of boiler steam which is required for filling these spaces each stroke.

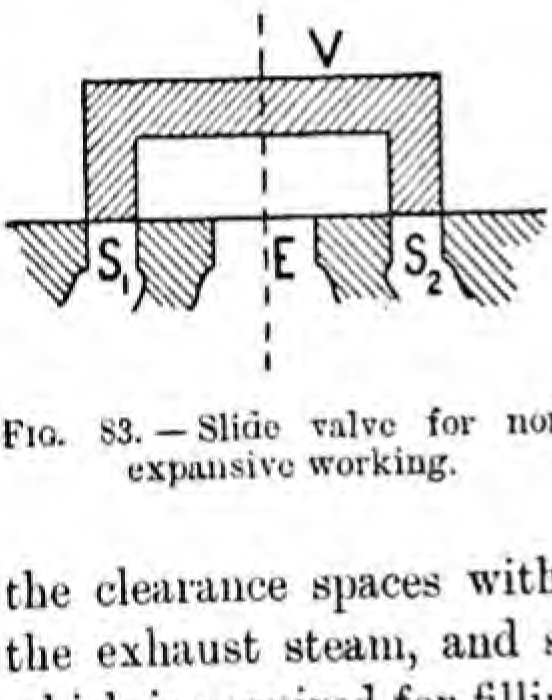


Fig. 83.—Slide valve for non-expansive working.

The student will find it helpful to study these events on a good working valve model; most laboratories possess such models. Failing this, cardboard models can be constructed easily by the student himself, or the excellent set of models designed by Messrs. Jones[1] may be used.

Slide valve for non-expansive working.—In Fig. 83, S_1 and S_2 are the steam ports and E is the exhaust port. The slide valve V is shown in the middle of its travel. It will be noticed that its length is equal to the distance between the outer edges of S_1, S_2, and that the length of the exhaust port in the valve is equal to the

[1] *Working Models for Engineering Students*, T. Jones and T. G. Jones; The Technical Publishing Co., Manchester.

distance between the inner edges of S_1, S_2. Draw a circle (Fig. 84) of radius OC, to represent the crank pin path, and another with the same centre and radius OE, to represent the path of the eccentric centre. Assume that the rotation is clockwise.

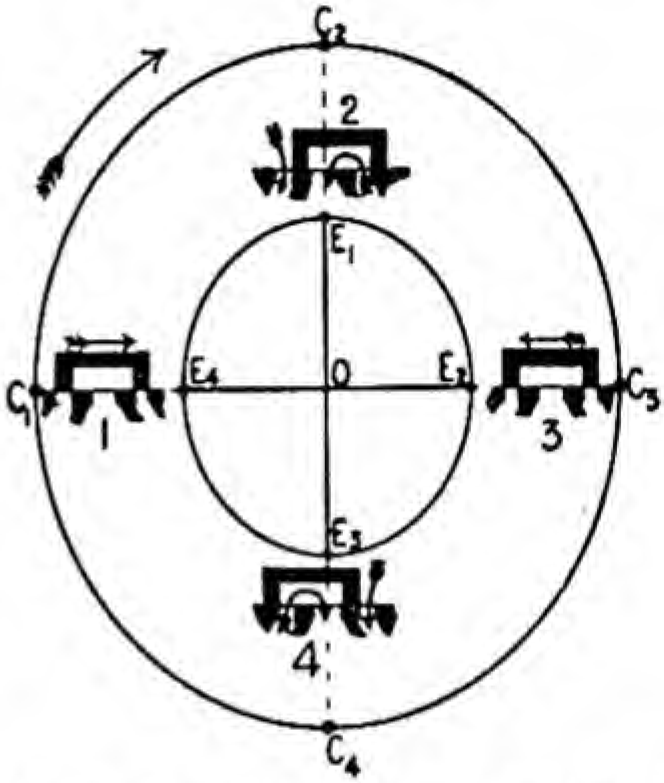


FIG. 84.—Diagram of valve and eccentric positions in non-expansive working.

Neglecting lead meanwhile, when the crank is at the dead point C_1, the valve must be in the middle of its travel and must be moving towards the right in order to open S_1 to steam. This is shown at 1 in Fig. 84. The position of the eccentric corresponding to this position of the valve will be E_1, OE_1 being at 90° to the crank. As both crank and eccentric are fixed to the shaft, their relative position will not be altered during rotation, *i.e.* the eccentric will always be 90° ahead of the crank.

When the eccentric reaches E_2, the port S_1 will be full open as at 2 (Fig. 84), and the crank will be at C_2. Steam will be cut off when the eccentric reaches E_3, the valve being then in the middle of its travel and moving towards the left as shown at 3; the crank will now be at the outer dead point C_3. Further rotation of the eccentric to E_4 will open S_2 to steam as shown at 4, the crank being at C_4.

Notice that steam is flowing into the cylinder through S_1, while the crank is moving from C_1 to C_3, *i.e.* throughout the whole forward stroke of the piston. On the backward stroke, steam will flow through S_2 into the cylinder while the crank is moving from C_3 to C_1. Notice also that the port S_2 is in communication with the exhaust throughout the forward stroke, and that S_1 is open to exhaust throughout the backward stroke. With this valve, therefore, there is neither expansive working nor cushioning.

EXPT. 23.—Test on your valve model a slide valve having neither lap nor lead. Note the crank angles at which the events of a complete revolution occur for each side of the piston.

Slide valve for expansive working.—To produce an earlier cut-off, the eccentric must be advanced through a small angle E_0OE_1 (Fig. 85). All the events will now take place earlier; thus, admission will take place as before when the eccentric is at E_0, but the crank will now be at C_0, the angle C_0OE_0 being made equal to the angle C_1OE_1, and not at C_1 as in the case dealt with above. Cut-off will take place when the eccentric is at E_2, the crank being then at C_2.

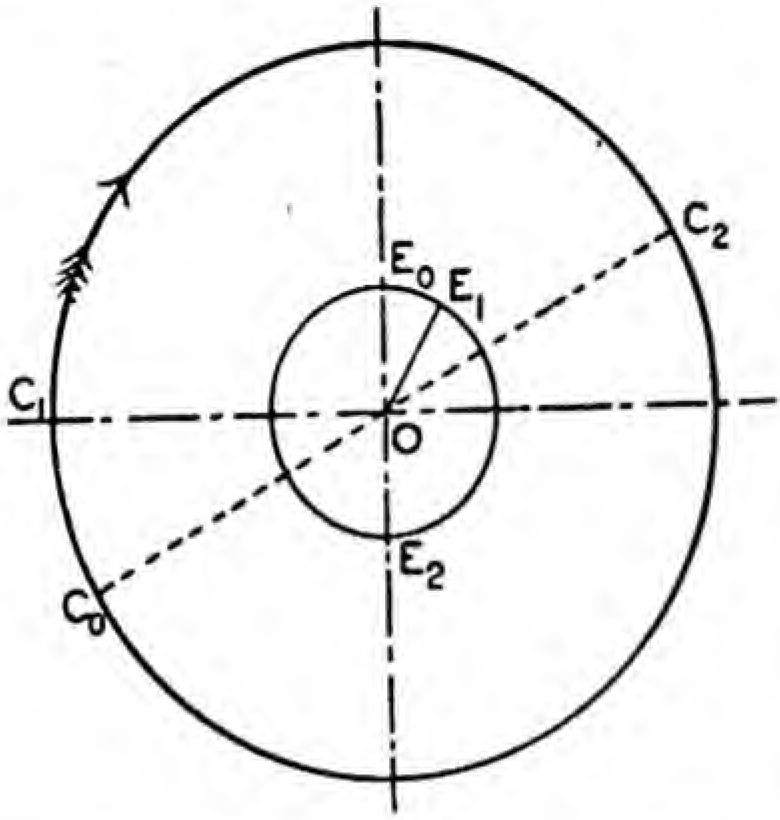


FIG. 85.—Advance of the eccentric to secure expansive working.

While this device of giving advance to the eccentric secures an earlier cut off, it also gives much too early admission, and the valve will have opened the steam port too much when the crank reaches the dead point. To correct this, add a piece to the length of the valve as shown in the valve section in Fig. 86. When the valve is

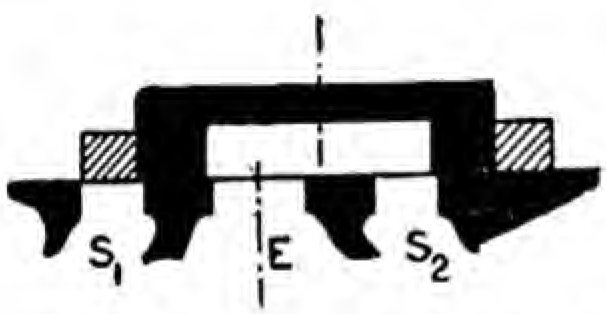


FIG. 86.—Lap in a slide valve for expansive working.

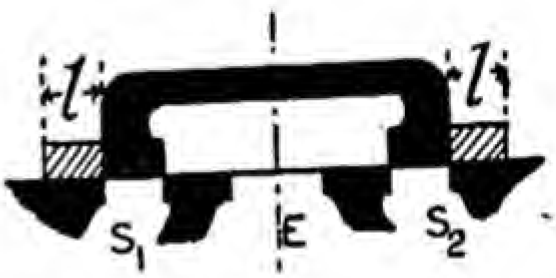


FIG. 87.—Measurement of lap.

brought to the middle of its travel, as in Fig. 87, its outer edges now project beyond the outer edges of the steam port. The distance between the outer edge of the valve, when in the latter position, and the outer edge of the steam port is called the **outside lap** of the valve.

We must therefore give advance to the eccentric and outside lap to the valve in order to secure expansive working.

Setting of crank and eccentric.—Taking a given valve as in Fig. 87, in which the outside lap is l, notice first the relation between the radius of the eccentric and the dimensions of the valve. The valve is shown in its mid-position in the diagram, and must be moved towards the right by an amount equal to l before admission occurs. Let a be the maximum amount of opening of S_1 to steam (usually this is a little less than the width of the steam port), then the valve must be moved a further distance equal to a in order to give this opening. As the same operations have to be repeated for the other steam port S_2, it follows that the travel of the valve must be equal to twice the lap plus twice the maximum opening to steam. Calling the radius of the eccentric r, this will give

$$r=l+a.$$

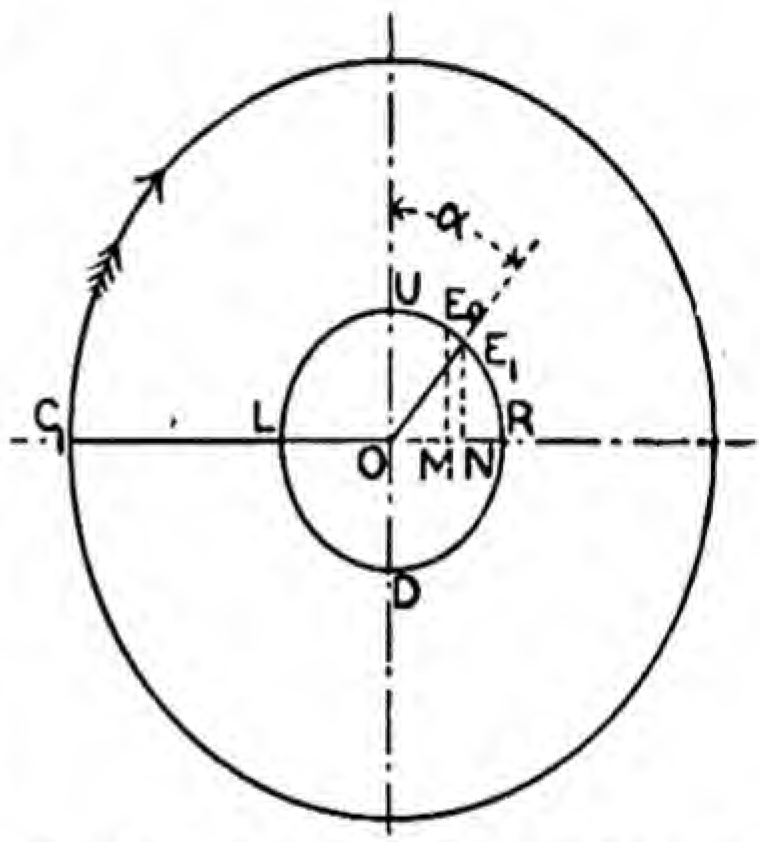


FIG. 88.—Setting of the eccentric for a slide valve having lap and lead.

Draw a circle of radius equal to r (Fig. 88) and mark the centre lines LR and UD on it. The dead points will be L and R. When the eccentric is at U, the valve will be in the middle of its travel as in Fig. 87. Make OM equal to the outside lap and draw ME_0 perpendicular to LR. This will give E_0 the eccentric position at admission. Make MN equal to the lead desired (*i.e.* the amount that the steam port is open to steam when the crank is at the dead point), and draw NE_1 perpendicular to LR. This will give E_1, the eccentric position when the crank is at the dead point C_1. The eccentric is therefore in advance of its 90° position by an angle $UOE_1=\alpha$. This angle is called the **angle of advance**. The crank and eccentric will be continually separated by the angle C_1OE_1.

Distribution of the steam.—The manner in which the valve distributes the steam to and from the cylinder can now be studied. In Fig. 89 is shown again the eccentric circle and the eccentric position E_1 copied from Fig. 88. Hitherto we have drawn a

separate circle to represent the crank, but if the eccentric circle be drawn full size, the same circle may be taken to represent the

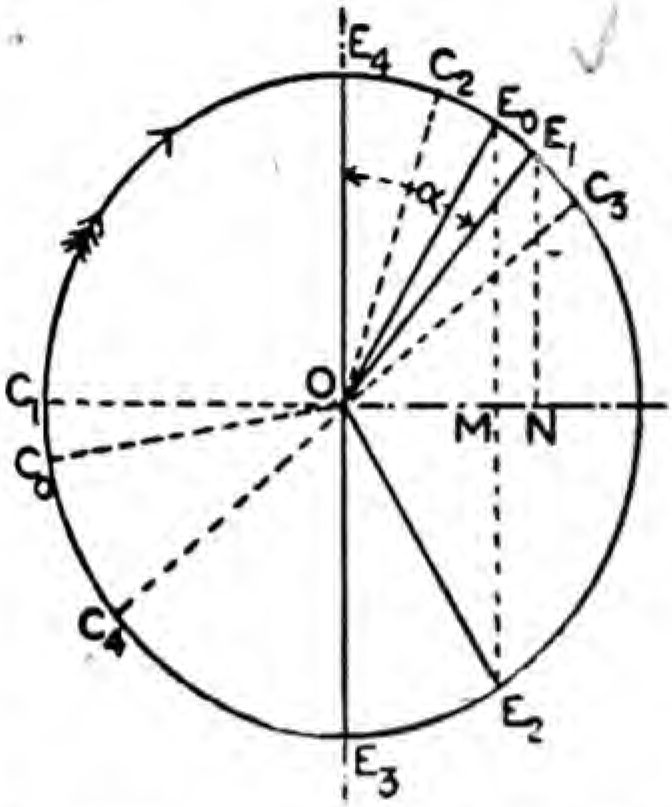


FIG. 89.—Events for the left-hand side of the piston.

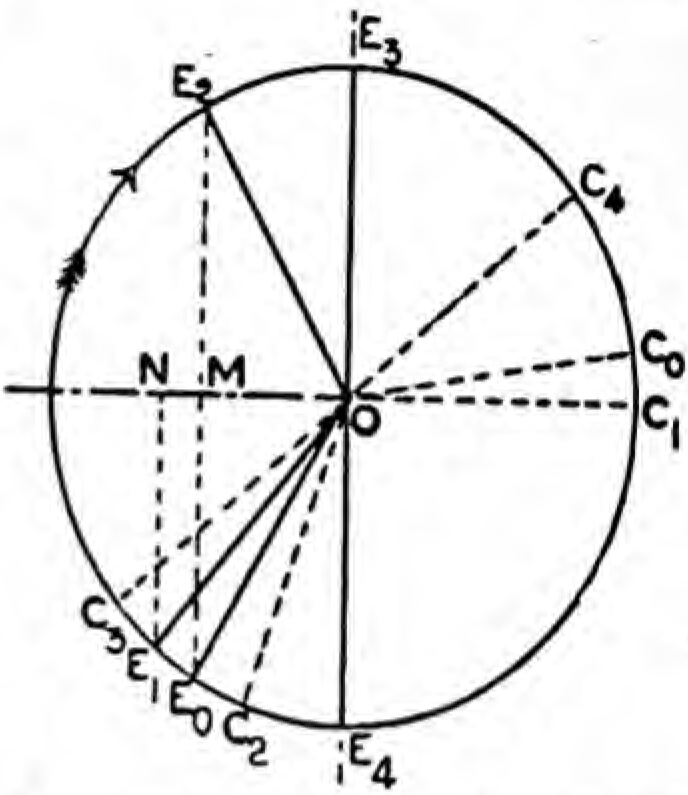


FIG. 91.—Events for the right-hand side of the piston.

crank to some other scale. Adopting this plan, the crank position corresponding to E_1 will be C_1. Mark again E_0, the eccentric at admission, and make the angle E_0OC_0 equal to the angle E_1OC_1. This gives C_0, the crank position at admission. Produce E_0M to cut the circle at E_2. When the eccentric is at E_2, the valve will be moving towards the left and will be a distance $OM=l$ from its mid-position; cut-off will therefore be occurring. Make the angle E_2OC_2 equal to the angle E_1OC_1, giving C_2, the position of the crank at cut-off. The positions of the valve at admission, dead point, and cut-off are shown at O, 1 and 2 respectively in Fig. 90. At release the valve, if its inner edges are dimensioned as in Fig. 87, will be in the middle of its travel and will be moving towards the left. The eccentric will therefore be at E_3 (Fig. 89), and the crank will be at C_3. At cushioning, the valve will be again in its mid-position and will be

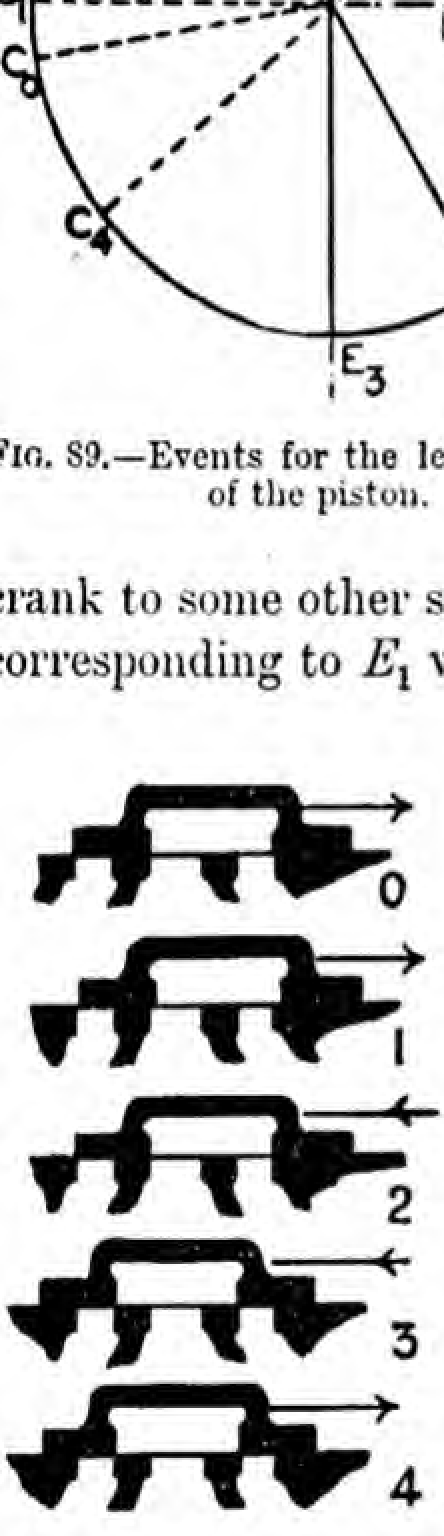


FIG. 90.—Valve positions for the events of the left-hand side of the piston.

moving towards the right so as to close S_1 to exhaust. The eccentric will be at E_4 and the crank at C_4. In each of these cases the crank is found in position by setting back from the eccentric position an angle equal to E_1OC_1. The valve at release and cushioning is shown in Fig. 90, at 3 and 4 respectively. The instants at which the events for the right-hand side of the piston occur will be found in the same manner and are shown separately in Fig. 91.

EXPT. 24.—Test on your valve model a slide valve having outside lap and lead, but no inside lap. Note the crank angles at which the events of a complete revolution occur for each side of the piston.

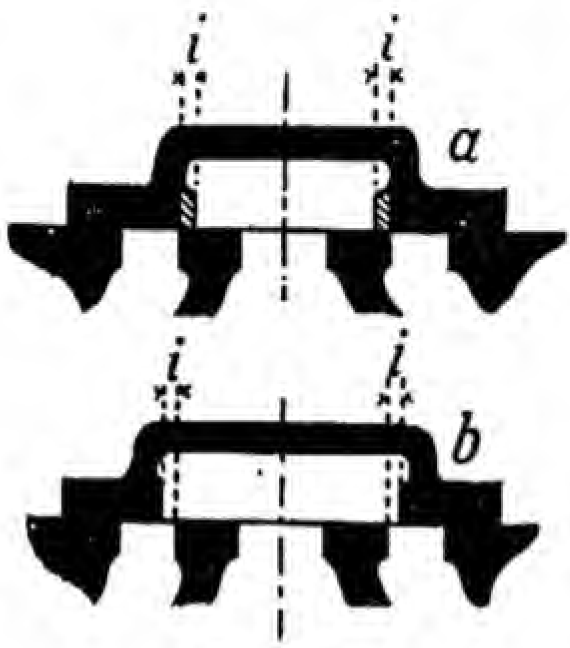


FIG. 92.—
(a) Valve having positive inside lap.
(b) Valve having negative inside lap.

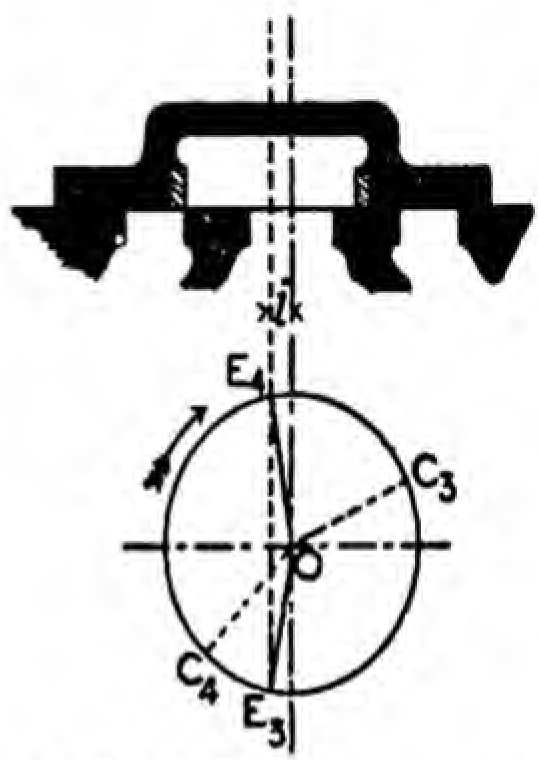


FIG. 93.—Setting of the eccentric for a valve having positive inside lap.

Inside lap.—It has just been seen that, if the inner edges of the valve are opposite the inner edges of the steam ports when the valve is placed in its mid-position (Fig. 87), the points of release and cushioning will occur when the eccentric is at 270° and 90° respectively from the inner dead point. To obtain later or earlier release or cushioning, a piece may be added to, or taken away from, the inner edge of the valve. If the piece be added, it is called **positive inside lap**, or, if taken away, **negative inside lap**. These are shown in Figs. 92 *a* and 92 *b*. Supposing a valve to have positive inside lap, release will occur when the inner edge of the valve is over the inner edge of the steam port and is moving towards the left (Fig. 93). The valve will be a distance i equal

to the positive inside lap from the mid-position, and the corresponding eccentric position will be E_3, the crank being an angle $E_3OC_3 = E_1OC_1$ behind. Release therefore occurs later. Cushioning will occur when the valve is again in the same position, but moving towards the right. The eccentric will therefore be at E_4 and the crank at C_4. Cushioning occurs earlier now. **The effect of positive inside lap is therefore to give later release and earlier cushioning.**

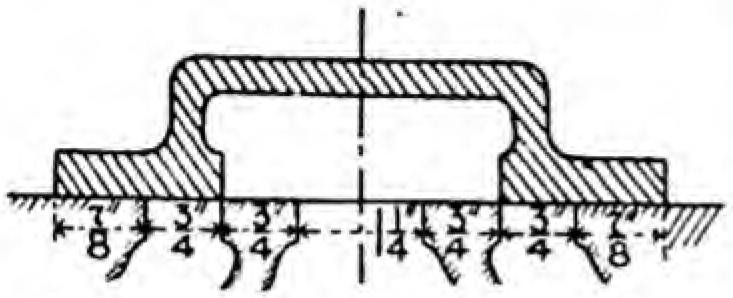


FIG. 94.—An example of a slide valve.

If the inside lap be **negative** then E_3 and E_4 will fall on the right of the vertical centre line by an amount equal to i. **Release will thus occur earlier and cushioning later.**

EXPT. 25.—Test on your valve model a slide valve having (*a*) positive inside lap, (*b*) negative inside lap. Note the crank angles in each case at which release and cushioning occur.

Piston position on diagram.—Taking the length of the horizontal diameter of the eccentric circle to represent the stroke of the piston to scale, the piston position corresponding to any of the crank positions may be marked on this diameter by the construction shown in Fig. 81. This will be carried out in the following example:

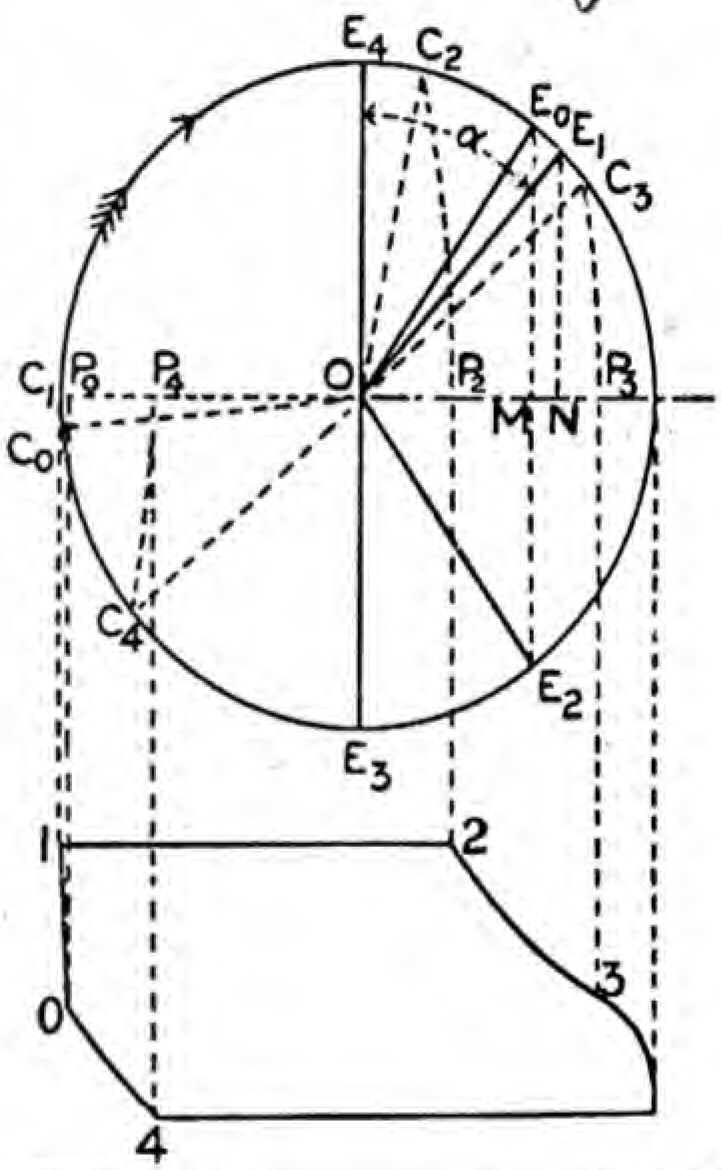


FIG. 95.—Construction for determining the probable indicator diagram for a given slide valve.

EXAMPLE. The ports and slide valve of a cylinder are shown in section in Fig. 94. The outside lap is $\frac{7}{8}''$, and the inside lap zero. The

travel of the valve is $2\frac{3}{4}''$ and the lead is $\frac{1}{8}''$. Draw a diagram showing the events for the left-hand side of the piston. Mark the piston positions and sketch a probable indicator diagram.

The answer is shown in Fig. 95. The construction followed is that described above and the expansion and cushioning curves are simply sketched in.

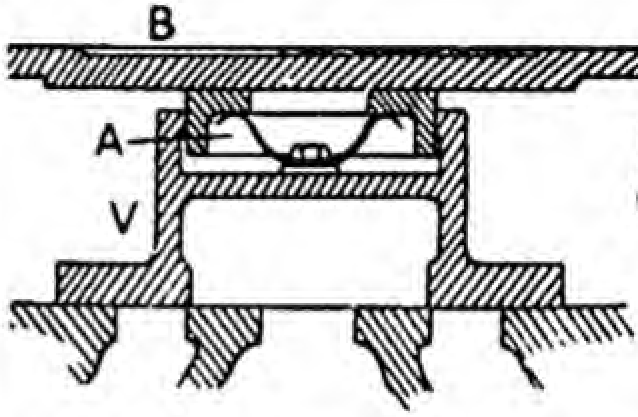


FIG. 96.—Slide valve having a relief frame.

EXPT. 26.—Test on your valve model the accuracy of your drawing in the above example. Note the piston positions at which the events occur for each side of the piston.

Balanced valves. — When the slide valve becomes very large, or is subjected to high steam pressure, the frictional resistances to its movements become very great. A considerable portion of the power developed by the engine is absorbed in driving such a valve, and the connecting parts must be strongly constructed. To obviate this waste of power, **relief frames** are sometimes fitted to the back of the valve so as to bear on the steam chest cover and thus relieve the valve of some of the steam pressure. Such a valve is shown in Fig. 96.

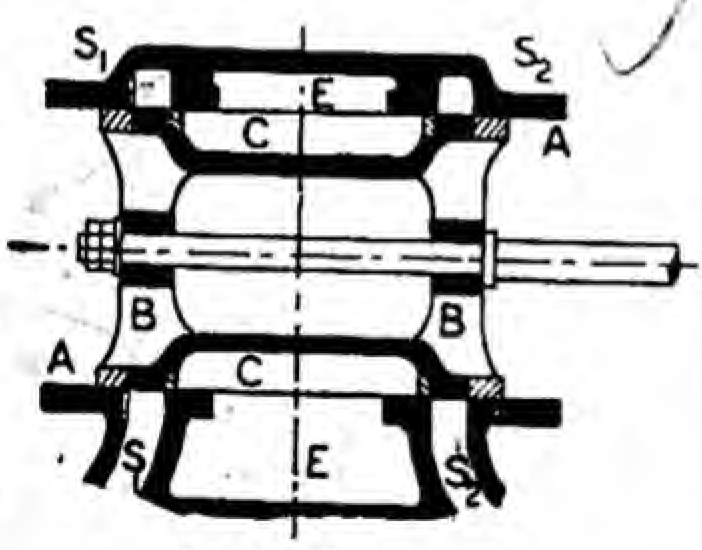


FIG. 97.—Piston valve.

In cases of engines working under high steam pressure, the **piston valve** is often adopted. In Fig. 97, S_1, S_2 are steam ports leading to the cylinder and opening into another cylinder A in which the valve works. E is the exhaust port also opening into A. The valve consists of a hollow cylinder turned to fit A at BB and reduced in diameter at C. The pieces BB serve as pistons, and uncover S_1, S_2 to steam and exhaust at the proper instants. The motion of the valve and the distribution of the steam are precisely the same as in the common slide valve, but the steam pressure has no influence in forcing the piston valve against the walls of the cylinder A and consequently the valve is driven easily.

Double-ported slide valve.—This valve has for its object the giving of a large opening to steam and exhaust with a comparatively small travel of the valve. In Fig. 98 *a*, S_1, S_2 are the steam ports, each having two openings at the valve face. E is the exhaust port. Passages A and B open through the sides of the valve into the steam chest. The passage C opens into the exhaust port. Fig. 98 *b* shows the valve opening S_1 to steam and S_2 to exhaust. Steam is flowing from the steam chest direct into S_1 and also through the passage A into the same port. On the other side, S_2 is open at two places to exhaust. A double opening is thus secured with this valve as compared with the opening which would be obtained with a common slide valve having the same travel. The setting of the eccentric, laps, etc., are identical with those of the common slide valve.

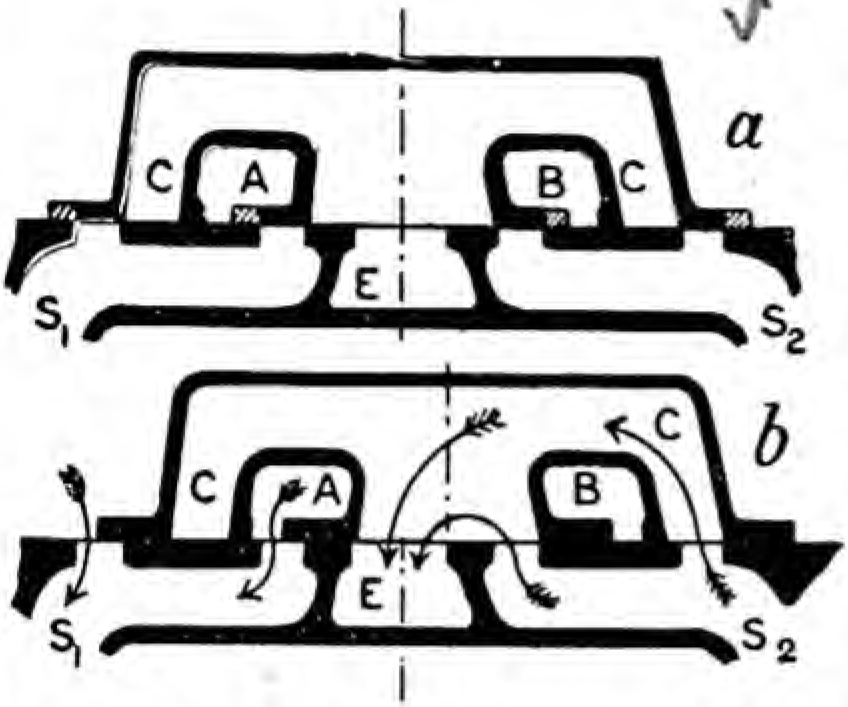


FIG. 98.—Distribution of steam by means of a double-ported slide valve.

Trick valve.—This valve has been designed to fulfil the same purpose as the double-ported valve so far as steam admission is concerned. The cylinder has three openings to the steam chest as in the common slide valve, S_1, S_2, E (Fig. 99 *a*). The valve has an internal passage A round which steam may pass. Its exhaust passage B is as in the common slide valve. In Fig. 99 *b*, the valve is seen admitting steam to S_1 direct from the steam chest and also through the passage A. S_2 is in communication through B with the exhaust. A

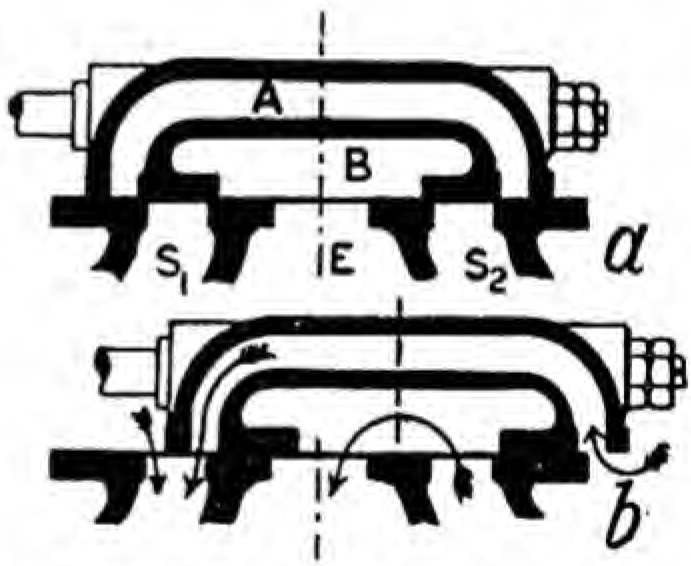


FIG. 99.—Distribution of steam by means of a Trick valve.

double opening to steam is thus secured for a given travel of the valve.

Link motions.—Link motions have for their principal object the reversal of the direction of rotation of the engine. Two eccentrics are employed and are set as shown in Fig. 100. An arrangement of links and rods is introduced whereby either eccentric may be used to drive the slide valve. Should E_1 be in gear with the valve, clockwise rotation will occur, and anti-clockwise rotation will be produced when E_2 gears with the valve.

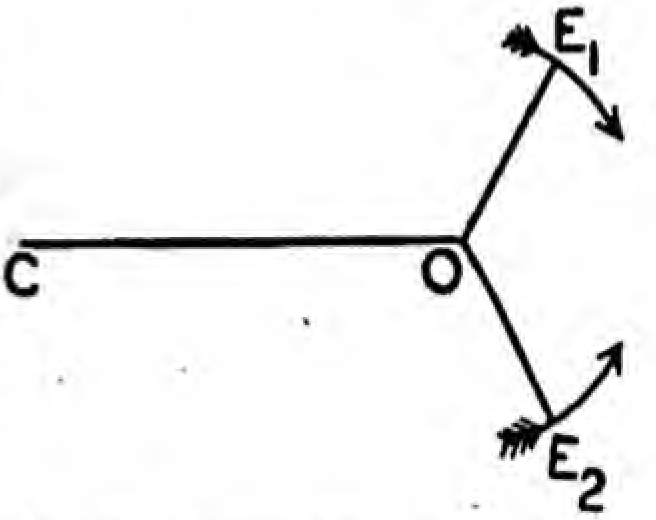


FIG. 100.—Setting of two eccentrics employed in link motions.

In **Stephenson's link motion**, an outline diagram of which is shown in Fig. 101, a curved link has its ends A and B connected to the eccentric rods E_1A, E_2B.

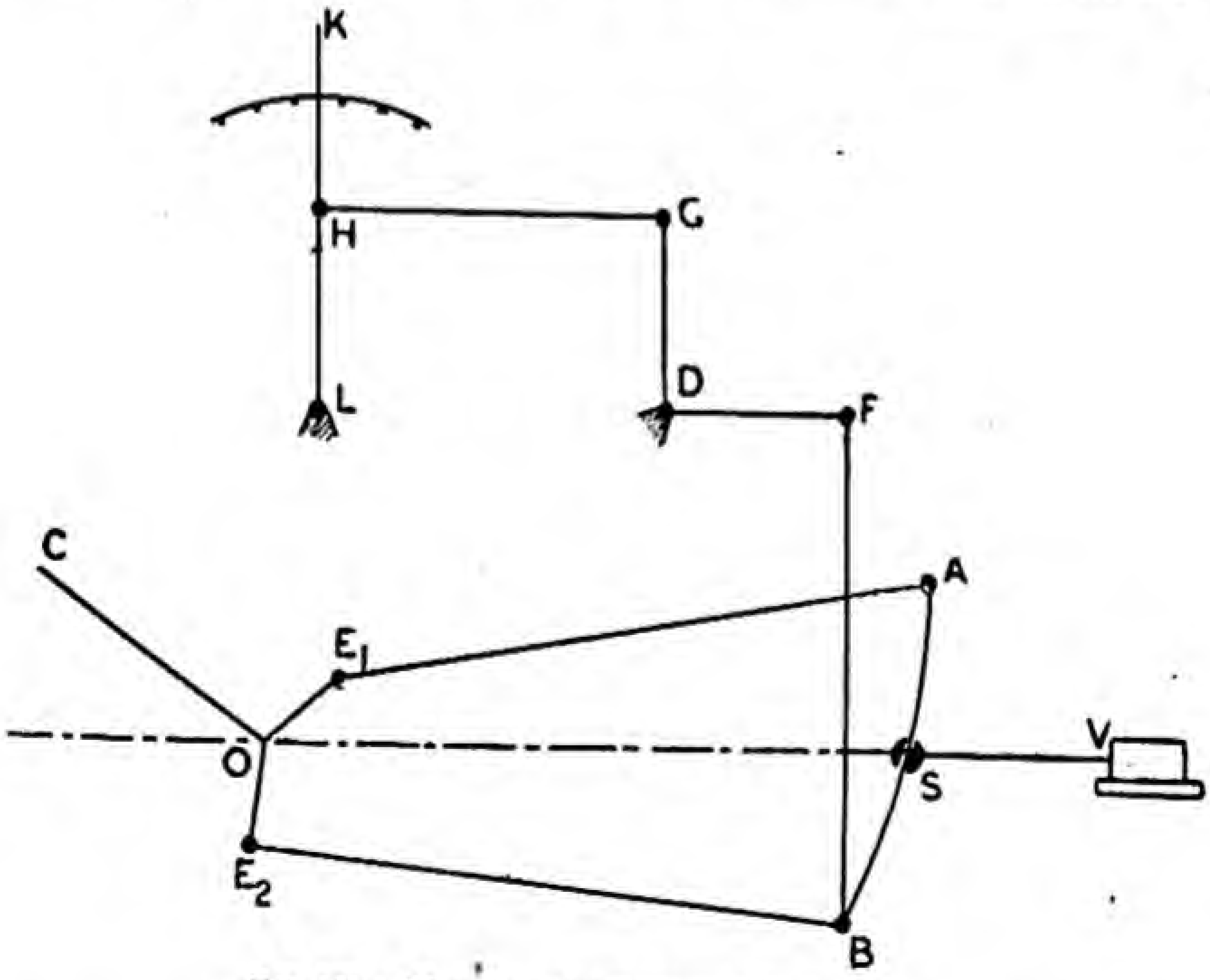


FIG. 101.—Diagram of Stephenson's link motion.

SV is the valve rod, engaging with the link at S by means of a block which may slide in the link. GDF is a bent lever pivoted

at D, and connected to the link AB by a rod FB. The bent lever is operated by means of a hand lever KL and rod HG, and when this is done, the link AB will be raised or lowered until

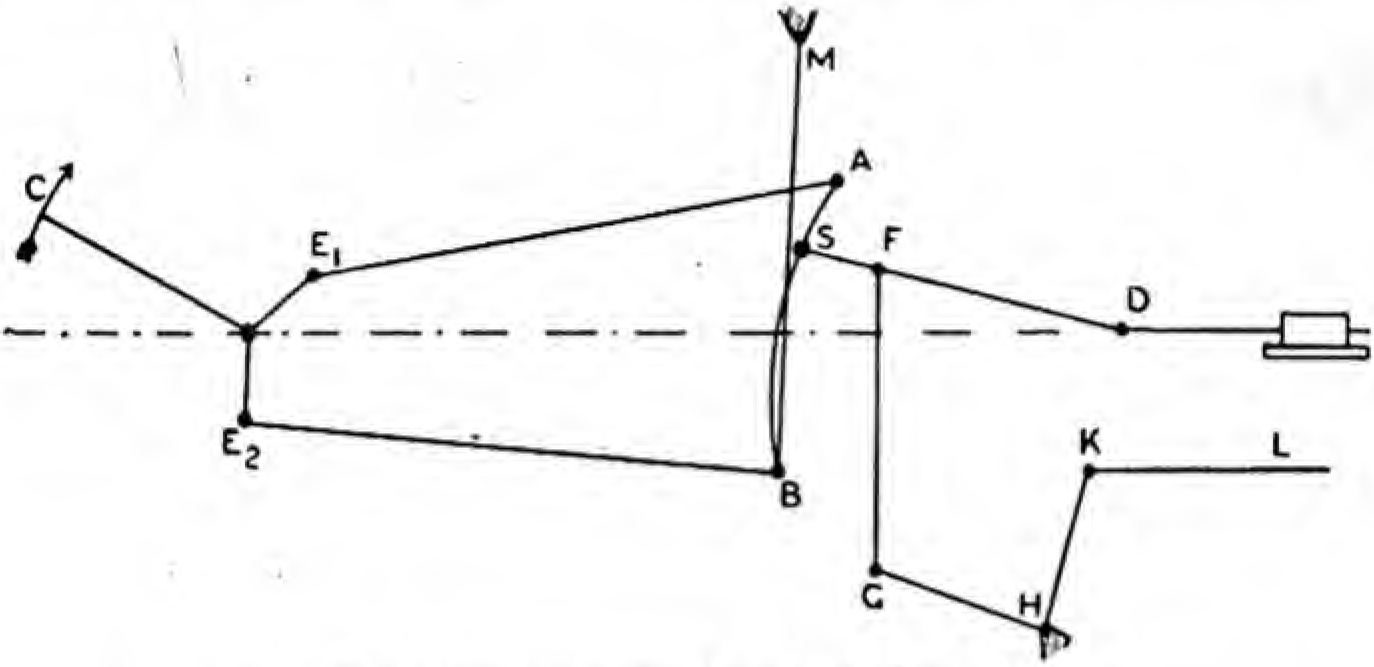


Fig. 102.—Diagram of Gooch's link motion.

either B or A coincides with S. In the former case, E_2 will drive the valve direct, the other eccentric meanwhile causing the link AB to vibrate without communicating any of its motion to the valve. The radius of the link AB is equal to the length of the eccentric rod or nearly so. Details of construction of this link motion will be found in Chaps. XV. and XVI.

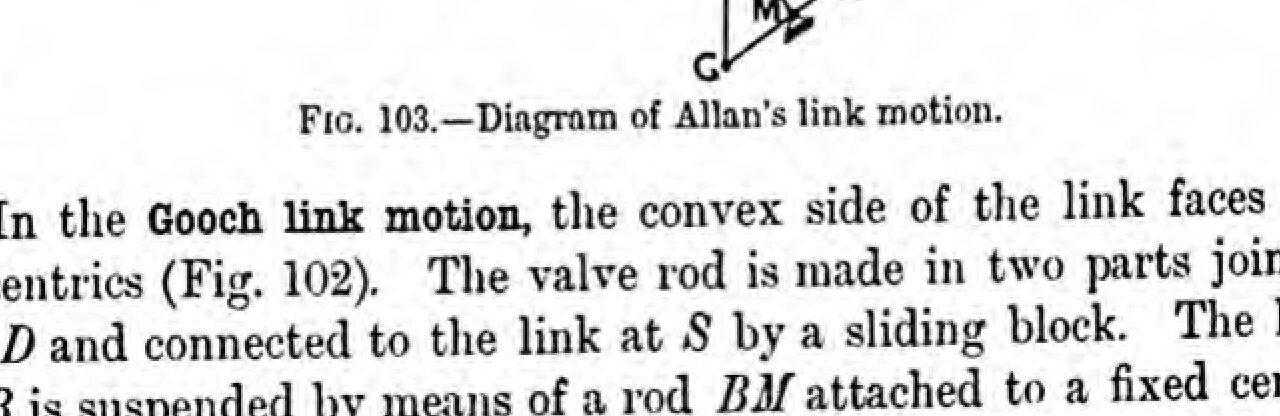


Fig. 103.—Diagram of Allan's link motion.

In the **Gooch link motion**, the convex side of the link faces the eccentrics (Fig. 102). The valve rod is made in two parts jointed at D and connected to the link at S by a sliding block. The link AB is suspended by means of a rod BM attached to a fixed centre at M, and is not raised or lowered. Reversal of the motion is accomplished by raising or lowering the rod SD, a bent lever GHK and rod GF being provided for this purpose. KL is a rod leading to the hand lever.

The **Allan link motion** is a combination of the Stephenson and the Gooch link motions. In Fig. 103, AB is the link and is made straight. The link is suspended by a rod GF connected to a T lever $GMLK$ by means of which it may be raised or lowered. The valve rod is in two pieces as in the Gooch gear, and the portion SD is suspended by the rod HK which is connected at K to the T lever. Movement of the T lever by pulling the rod LN towards the right will raise the link AB and lower the valve rod SD.

Variable cut-off in link motions.—By bringing the sliding block S in any of the above link motions to some intermediate point in the link AB, both eccentrics will contribute to the motion of the valve. The resulting motion is best studied by aid of a

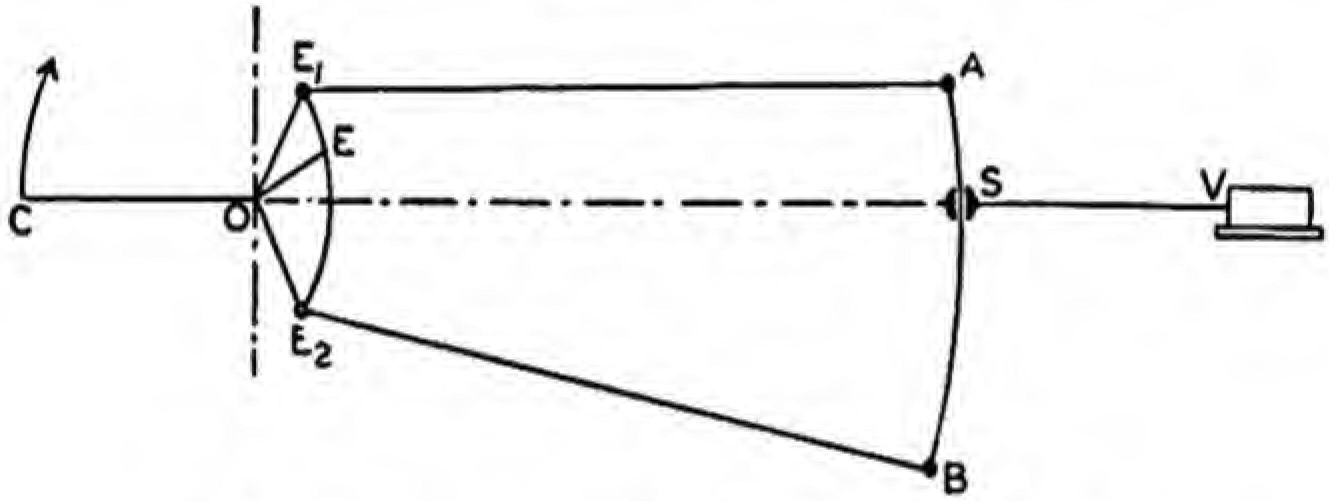


FIG. 104.—Equivalent eccentric in Stephenson's link motion.

model, such as is to be found in most laboratories, or the link may be drawn in several different positions, and the position of the valve corresponding to any given crank angle will be thus found. An approximate solution is due to Mr. Macfarlane Gray, and consists in finding the radius and setting of an **equivalent eccentric**, which, if connected direct to the valve, would give to it nearly the same motion as that which it derives from the actual eccentrics. Fig. 104 shows the method applied to a Stephenson's link motion. Connect the eccentric centres with a circular arc of radius found from

$$\text{Radius} = \frac{E_1E_2 \times E_1A}{2AB}.$$

Divide the arc E_1E_2 at E in the same proportion that the point S divides the link AB. Thus

$$E_1E : EE_2 = AS : SB.$$

Then OE is the equivalent eccentric. Notice that it is of shorter radius than the actual eccentrics and has a greater angle of

advance. The general effects of "notching up" the link will therefore be to increase the lead, to diminish the travel of the valve and consequently the opening to steam, and to cause all the events to occur earlier. Should the eccentric rods be crossed for the given position of the crank OC, the construction differs from that given only in making the arc E_1EE_2 convex towards the centre O. Crossed rods cause the travel of the valve and also the lead to diminish very much on notching up the link; the arrangement is seldom used.

EXPT. 27.—Test on your valve model a given valve operated by Stephenson's link motion. Note the crank angles at which the events occur for different settings of the block in the link.

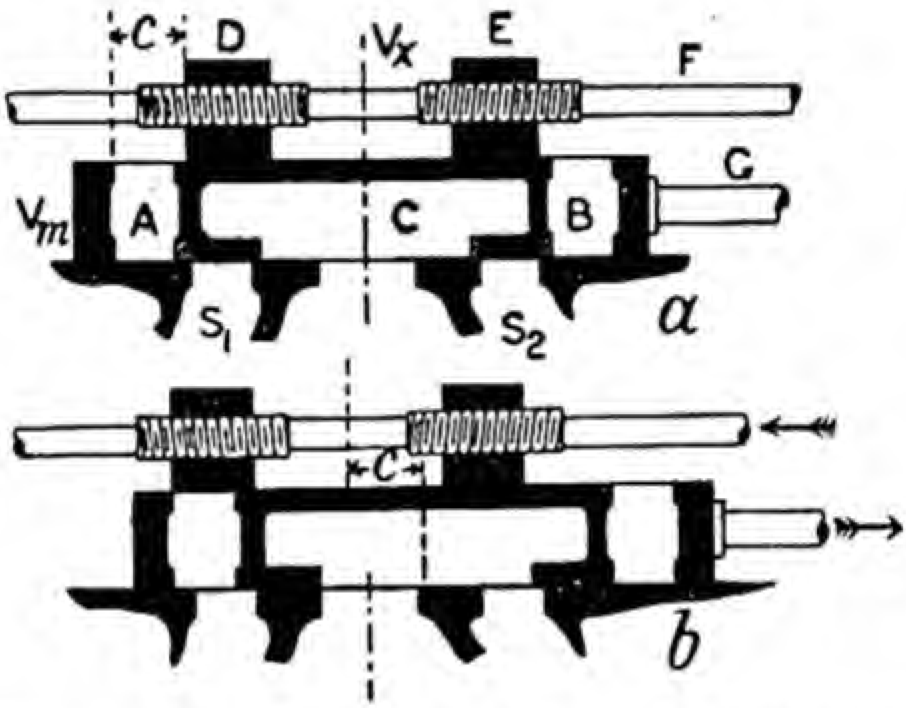


FIG. 105.—Meyer's valve gear for securing a variable cut-off.

Meyer's valve gear.—In Meyer's gear there are two slide valves, one working on the back of the other. In Fig. 105 a, V_m is a slide valve made of extra length and having the usual exhaust port C. A and B are two ports passing through the valve; steam enters the cylinder ports S_1, S_2 after passing through A and B. V_x is another valve working on the back of V_m, and consisting of two blocks D and E screwed to fit the right and left handed threads on the valve rod F. Rotation of the valve rod F will cause the blocks D and E to approach or recede from one another, depending on the direction of rotation, and by this means the object of the gear is fulfilled, viz. to obtain later or earlier cut-off. V_m is called the **main valve** and V_x the **expansion valve.**

There are two eccentrics, one for driving each valve. The main eccentric driving V_m is set as for a common slide valve, and the main valve is given sufficient lap to cause it to cut off at the latest point required. Release and cushioning are controlled entirely by the port C in the main valve.

The expansion valve serves the sole purpose of effecting an early cut-off. The eccentric driving it is usually of the same radius as that driving the main valve, so that both valves have the same travel. The expansion eccentric is generally set exactly opposite the crank, although this position may be varied slightly.

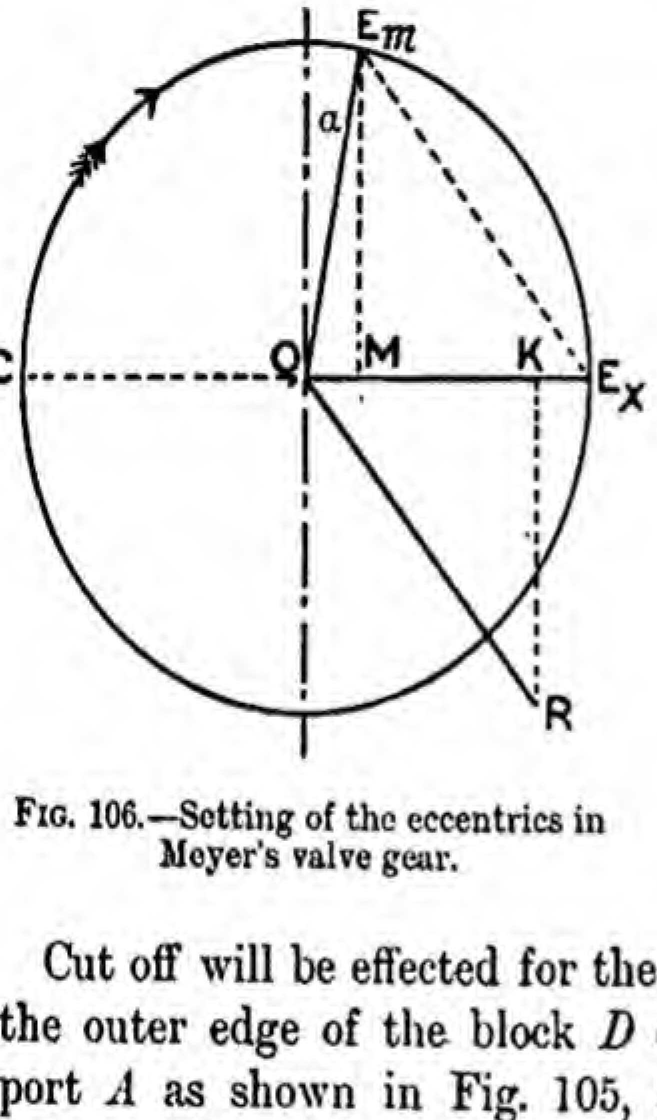


FIG. 106.—Setting of the eccentrics in Meyer's valve gear.

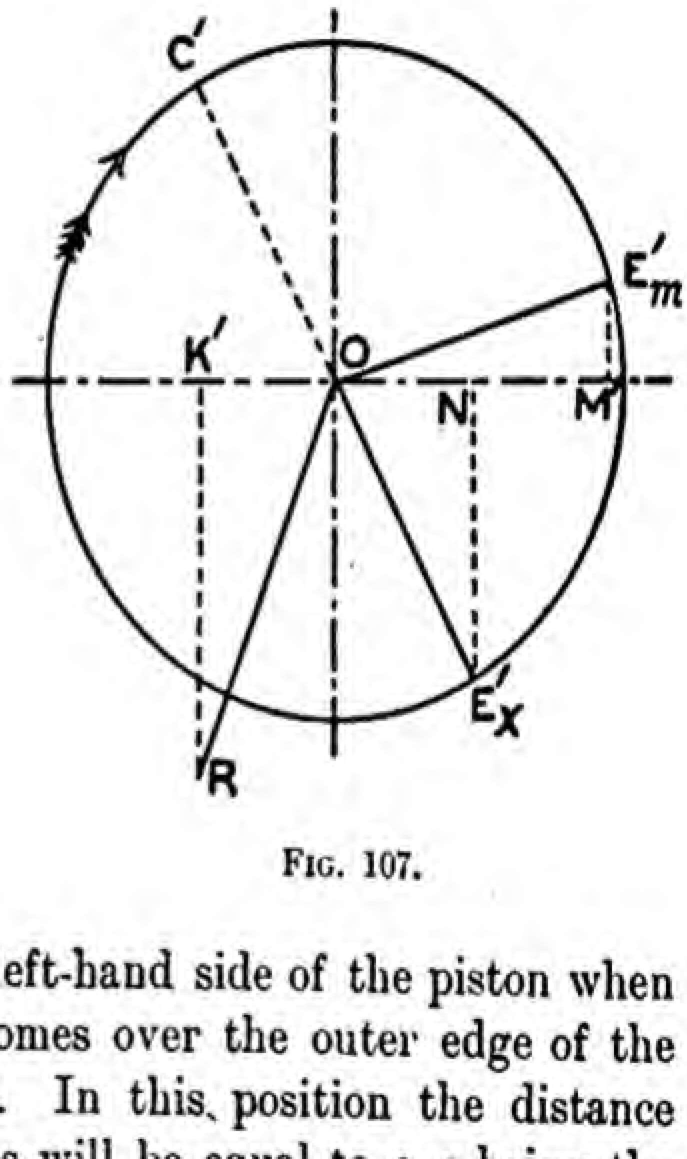


FIG. 107.

Cut off will be effected for the left-hand side of the piston when the outer edge of the block D comes over the outer edge of the port A as shown in Fig. 105, b. In this position the distance between the centres of the valves will be equal to c, c being the distance from the edge of D to the edge of A when both valves are brought to the middle of their travel.

The setting of the eccentrics is shown in Fig. 106. The crank is shown on the dead point. OM will be equal to the outside lap plus the lead of the main valve. E_m is the main eccentric and a its angle of advance. E_x is the expansion eccentric set opposite the crank.

In this diagram OM is the displacement of the main valve from its mid-position and OE_x is that of the expansion valve. The distance between the valve centres will therefore be equal to ME_x.

In Fig. 107 is shown the crank and eccentrics in any other given position. OM' and ON will be the displacements from mid-positions of the main valve and the expansion valve respectively, and therefore NM' is the distance between their centres. When NM' becomes equal to c in Fig. 105, cut-off occurs.

A convenient construction for giving the distance between the valve centres for any crank position is as follows. Join E_mE_x in Fig. 106, and draw OR equal and parallel to E_mE_x. Draw RK perpendicular to CO. Then the triangles ORK and OE_mE_x are equal in all respects, and therefore OK is equal to ME_x and consequently shows the distance between the valve centres. Carrying out the same construction in Fig. 107, it will be easily seen that OK' is equal to NM' and is therefore again equal to the distance between the valve centres. OR may therefore be taken as a **resultant eccentric**, which will show, for any crank position, the distance between the valve centres. OR will rotate with the crank and will be in advance of it by the angle $C'OR$.

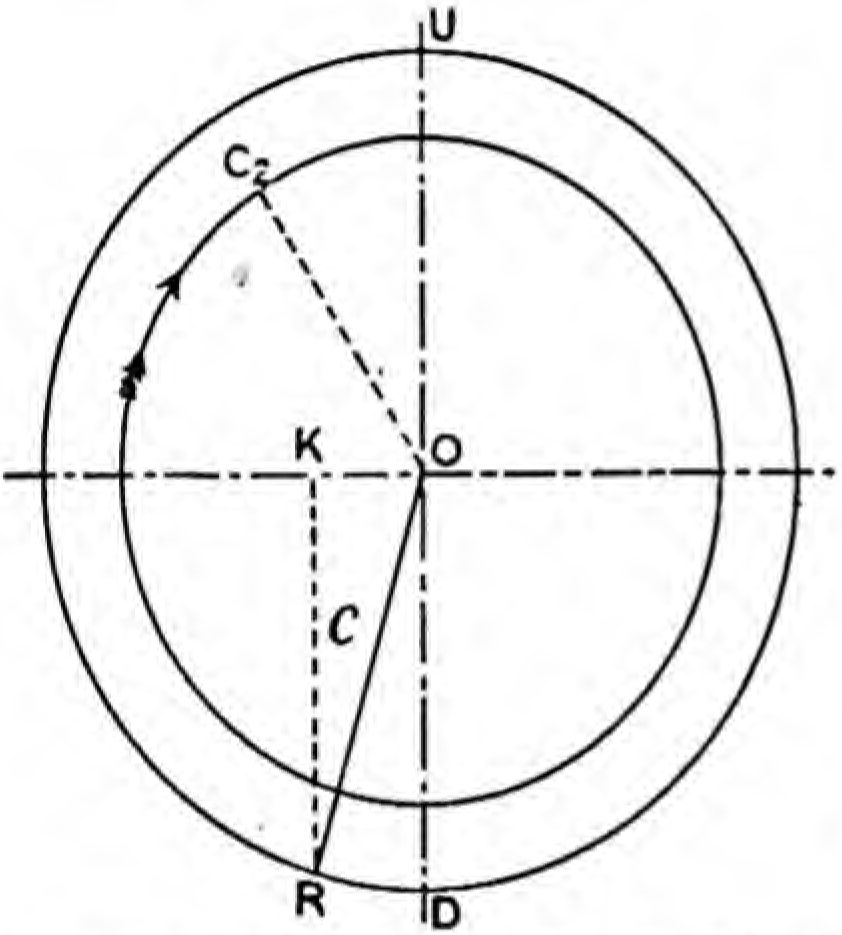


FIG. 108.—Resultant eccentric in Meyer's valve gear.

To find the position of the crank at cut-off for a given value of c in Fig. 105, draw a circle to represent the paths of E_m and E_x and another to represent the path of R (Fig. 108). Make OK equal to c, and draw KR parallel to UD. This will give the position of OR at cut off. Set back the angle ROC_2, taken from Fig. 106, thus giving OC_2, the crank at cut-off. It will be noticed that the smaller the value of c is, the earlier will be the cut-off, *i.e.* separating the blocks of the expansion valve gives earlier cut-off, and bringing them closer together gives later cut-off.

EXERCISES ON CHAPTER VIII.

1. The connecting rod of an engine is $4\frac{1}{2}$ times the length of the crank. Find, by drawing, the fraction of the stroke accomplished by the piston when the crank has rotated 30°, 60°, 90°, 120°, 150°, from the inner dead point. Make a table of your results.

2. Explain how the slide valve for non-expansive working may be modified in order to produce expansive working. What alteration must be made in the eccentric setting? Confine your answer to the points of admission and cut-off.

3. The travel of a slide valve is $2\frac{1}{2}''$, the outside lap is $\frac{1}{2}''$ and the lead is $\frac{1}{8}''$. Find the angle of advance, and the crank positions at admission and cut-off.

4. In question 3, suppose that there is no inside lap and find the crank positions at release and cushioning.

5. Take the results obtained in questions 3 and 4, and draw the hypothetical diagram. Assume that the connecting rod is $4\frac{1}{2}$ times the length of the crank, the admission pressure 80 lbs. per square inch by gauge, and that the engine is non-condensing.

6. Give sketches and description of a piston valve, or of a double-ported slide valve.

7. Sketch and describe a slide valve having a relief frame.

8. Explain by reference to an outline diagram the construction and action of Stephenson's link motion.

9. Answer question 8 for the case of crossed eccentric rods. What changes will this arrangement effect in the cycle when the gear is "notched up"?

10. Give sketches and a description of Meyer's valve gear. Explain how the cut-off is affected by altering the positions of the expansion blocks.

11. Draw a slide valve in its mid position, and in dotted lines show its position at the beginning of the stroke of the piston. Explain how it distributes steam. What do we mean by half-travel, angular advance and lap of a valve? 1905.

12. A slide valve is worked directly from an eccentric. The advance is 30°. When the main crank has moved 20° from the line of centres, show the position of the eccentric crank. The half-travel of the valve being 3 inches, mark off this radius and drop a perpendicular on the line of centres; what have you thus found? 1906.

13. If steam is cut off both in the down and up strokes of a vertical engine (the crank below the cylinder) when the crank makes an angle

of 70° with a dead point; show that this means a later cut-off in the down stroke than in the up stroke. Is this a good or a bad result? 1900.

14. A link motion or other gear for a slide valve will reverse an engine, but suppose we do not reverse the engine; suppose we only change from say full to half gear, state clearly what it is that is really affected by the change. Sketch also the probable change in the indicator diagram. 1903.

CHAPTER IX.

MECHANICS OF THE ENGINE.

Turning moment.—The forces acting on the piston of an engine are transmitted through the piston rod and connecting rod, and so exert push, or pull, on the crank pin; these forces acting on the crank pin have a **tendency to rotate the crank shaft**, which

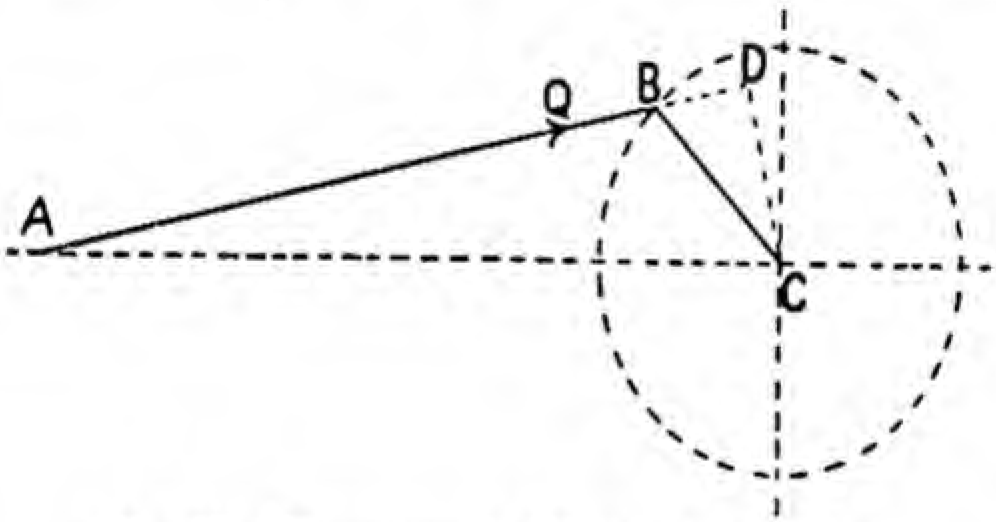


FIG. 109.—Measurement of the turning moment.

tendency is described as the **turning moment**. The moment of a force is measured by the product of the magnitude of the force and the length of the perpendicular dropped from the axis of rotation on to the line of action of the force. In Fig. 109, AB is the connecting rod and BC the crank. Q is a force of 15,000 lbs. acting along the connecting rod. To measure the turning moment of Q, drop a perpendicular CD from C, the axis of rotation, on to the line of the connecting rod, producing it, if necessary. Suppose CD to be 22 inches, then

$$\begin{aligned}\text{Turning moment} = T &= 15{,}000 \times 22 \\ &= 330{,}000 \text{ lb.-inches} \\ &= \underline{27{,}500} \text{ lb.-feet.}\end{aligned}$$

Forces acting at crosshead.—In general, there will be three forces acting at the crosshead pin, viz. the force P acting along the piston rod (Fig. 110, i), the force Q along the connecting rod, and a reaction S coming from the guides, these three forces being in equilibrium. If we neglect friction, S will always act at right angles to the line of the stroke. Assuming friction absent for our present purpose, the values of Q and S can be found, if we know P, by an application of the parallelogram of forces. Thus, set off Aa

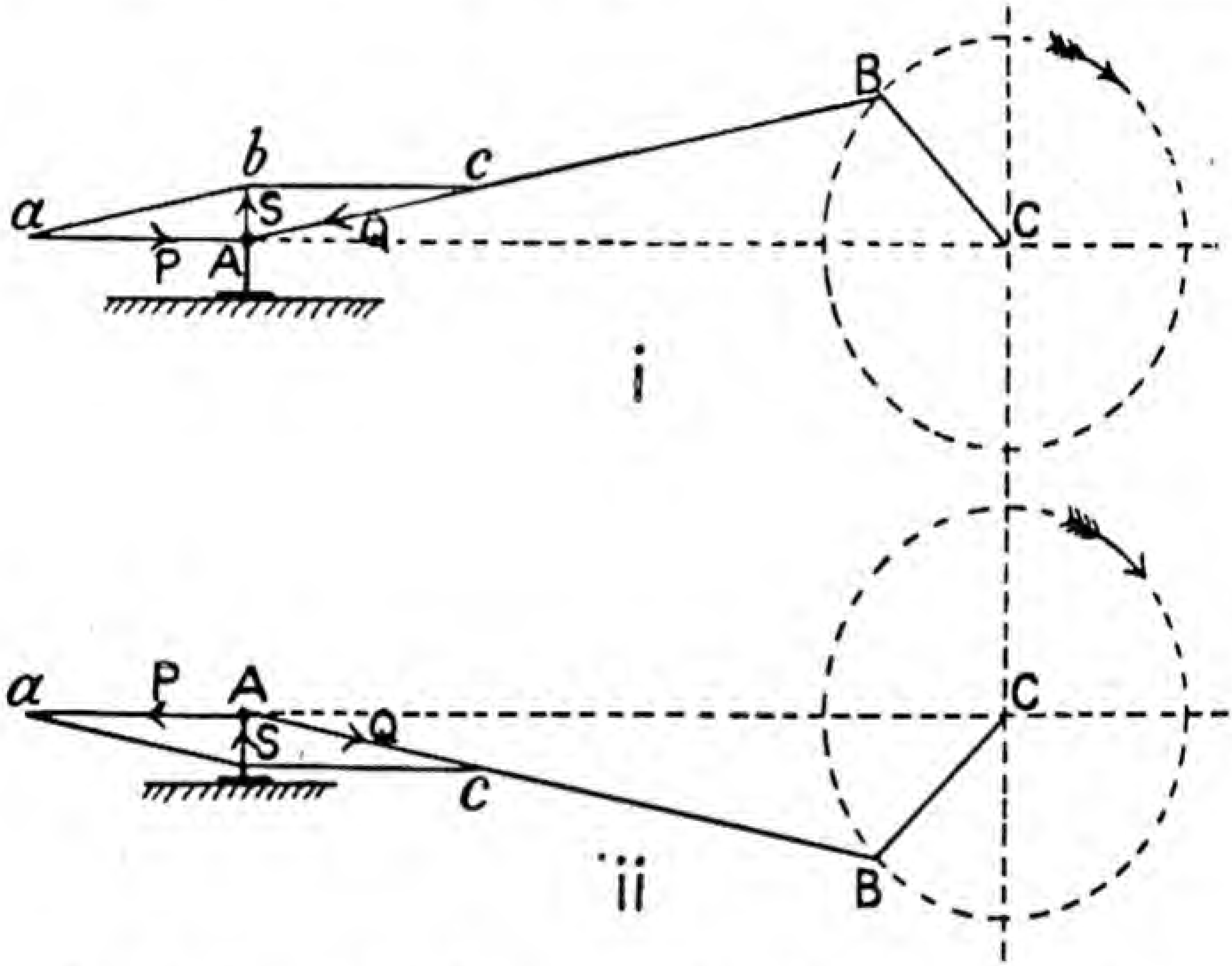


FIG. 110.—Diagram of the forces acting at the crosshead for a crank having clockwise rotation.

to represent P to a convenient scale of force, draw ab parallel to AB cutting the line of S in b; draw bc parallel to AC cutting the connecting rod in c. Then, to the scale of force, cA gives the force Q in the connecting rod and Ab gives S the reaction of the guide.

Fig. 110 ii shows the crank in another position in the return journey. P and Q now act in opposite directions, the piston and connecting rods being under pull. Notice that S, the reaction of the guide, still **acts upwards**. Suppose, however, we reverse the direction of rotation and draw diagrams as in Fig. 111, it will be found that the reaction of the guide is always a **downward force.**

This shows the necessity for designing the slipper so that both upward and downward reactions are suitably provided for in

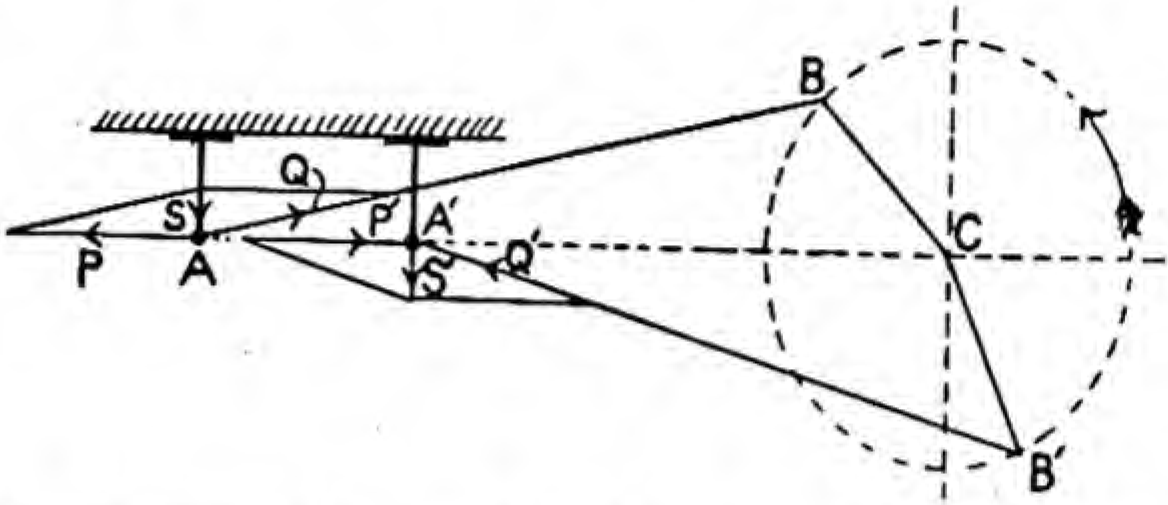


FIG. 111.—Diagram of the forces acting at the crosshead for a crank having anti-clockwise rotation.

engines which are intended to reverse (Fig. 112); and it is customary to do so in all engines as there is a liability for the direction of the reaction of the guide to be reversed when the crosshead is nearing the ends of its stroke.

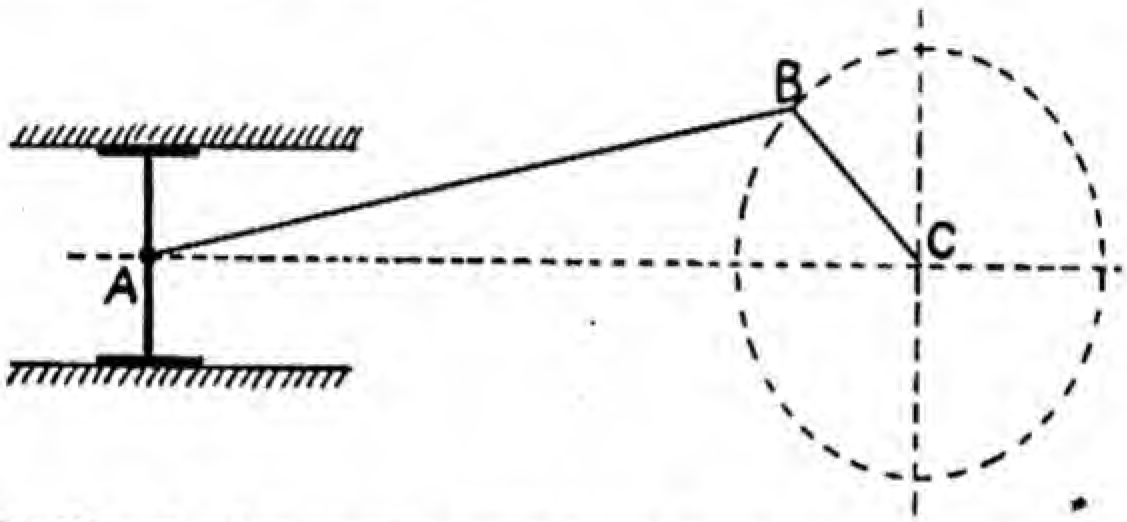


FIG. 112.—The slipper must be guided on both sides in engines intended to reverse.

Force acting on the piston.—To obtain the net force acting on the piston when the crank is at any given position, the following method may be employed. It is assumed that an indicator diagram for the engine under consideration is available, or that a diagram can be drawn which will as nearly as possible represent the true indicator diagram. Copy the diagram accurately on a sheet of drawing paper by pricking through (Fig. 113 i). Draw an outline diagram of the crank and connecting rod on a centre line AC, making ef, the travel of the crosshead, equal in length to AL, the length of the indicator diagram, e and f being projected from A

and L respectively. The crank radius BC will be one half of ef and the length of the connecting rod AB should be calculated from

$AB : BC$ = actual length of connecting rod : actual length of crank.

Examining the left-hand indicator diagram, which shows the pressures on the left-hand side of the piston throughout a revolution of the crank, it should be noted that the curve ab gives the

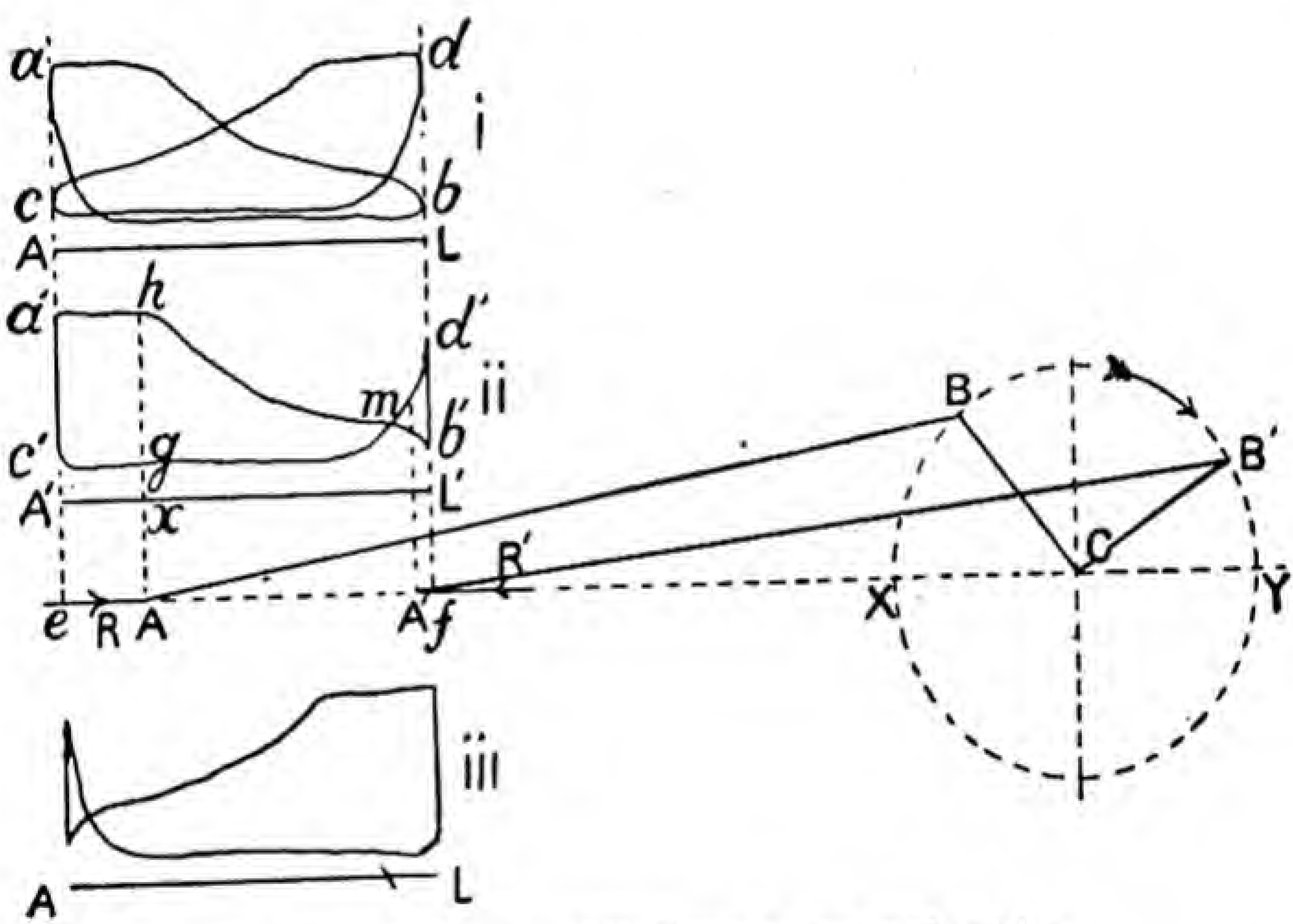


FIG. 113.—Diagram of the forces acting on the piston.

pressures on the left side of the piston during the left-to-right stroke, the remainder of the curve from b back to a gives the back pressure on the same side of the piston during the return stroke. The back pressure on the piston during the left-to-right stroke will be given by the curve cd belonging to the right-hand diagram. It is convenient to plot a diagram showing the net pressure on the piston. This is shown in Fig. 113 ii, immediately below the indicator diagram, and is obtained by copying the curves $a'b'$ and $c'd'$ from the curves ab and cd of the indicator diagram.

Draw the crank in any given position CB, say at 60° to the line of the stroke, and find the position A of the crosshead pin, which

may be taken to represent the piston position, the stroke of the piston being then regarded as from e to f. From A draw Agh perpendicular to AC, cutting the curves of the diagram in g and h, and the atmospheric line in x. The pressure per square inch, reckoned above atmospheric pressure, on the left-hand side of the piston, will be given by xh to the scale of pressure, and that on the other side of the piston will be given by xg. To obtain the total net pressure on the piston, it will be remembered that the full area of the piston is available on the left-hand side, and that the area of the piston rod must be deducted from the right-hand side.

Let D = diameter of piston, in inches.

d = diameter of piston rod, in inches.

R = total net pressure on piston.

Then

$$R=\left(\text{pressure } xh \times \frac{\pi D^2}{4}\right)-\left\{\text{pressure } xg\left(\frac{\pi D^2}{4}-\frac{\pi d^2}{4}\right)\right\}\text{lbs.}$$

For practical purposes it is often the custom to neglect the diminution in effective area of one side of the piston due to the rod. Accepting this approximation we may write

$$R=\text{pressure } gh \times \frac{\pi D^2}{4}\text{ lbs.}$$

It will be noticed that by this approximation the height of the diagram between the two curves gives the net resultant pressure per square inch on the piston at any instant. The resultant will act towards C for all parts of the stroke excepting that portion which lies to the right of the point m (Fig. 113 ii), where the curves cross. Within this latter portion the resultant acts away from C, as shown at R', the effect being due to the cushioning action.

The diagram for the return stroke is constructed in a similar manner and is shown immediately below the line of the stroke (Fig. 113 iii). Such diagrams are called **piston effort diagrams.**

Force acting in the piston rod.—If nothing were to interfere, the force with which the piston rod acts on the crosshead would be equal to the net resultant pressure on the piston. The friction of the piston rubbing in the cylinder and of the rod rubbing in the stuffing-box diminishes the available force. We may neglect the effect of friction for our present purpose and proceed to examine a more important matter. It is a well-known principle in

mechanics that **force is required to produce any change in the velocity of a body.** Thus, if a body is at rest and has to be set into motion, force is required, and a body which is in motion requires the application of a force in order to bring it to rest. In fact, all bodies possess **inertia,** which is the property of disinclination to alter the state of rest or of uniform velocity in a straight line. Force is required to overcome inertia. The force required to produce a given change of motion in a body is measured by

$$P = \frac{wa}{g} \text{ lbs.}$$

Where P = force required,
w = weight of body in lbs.
a = acceleration of body in feet per second per second.
g = acceleration due to gravity, which may be taken as 32·2 feet per sec. per sec.

The acceleration is measured by dividing the change in velocity in feet per second by the time in seconds in which the change took place.

Example. The piston of an engine weighs 140 lbs. and is brought from a state of rest to a velocity of 6 feet per second in 0·5 second. What has been the average acceleration? What average force must have acted on it?

Here the change in velocity is 6 feet per second, therefore

$$\text{Average acceleration} = \frac{\text{change in velocity}}{\text{time}}$$

$$= \frac{6}{0{\cdot}5} = \underline{12} \text{ feet per sec. per sec.}$$

$$\text{Average force,} = \frac{wa}{g}$$

$$= \frac{140 \times 12}{32{\cdot}2}$$

$$= \underline{52{\cdot}2} \text{ lbs.}$$

Effect of inertia.—The piston, piston rod, and crosshead, called the **reciprocating parts,** come to rest at each end of the stroke, and have different velocities at every part of the stroke, *i.e.* they undergo acceleration, and consequently force is required to give them the proper acceleration and so overcome their inertia.

EXPT. 28.—Hang a helical spring to a spring balance suitably supported on a stand (Fig. 114) and attach a weight to the lower end of the helical spring. Pull the weight down three or four inches and let go. The weight will vibrate up and down, coming to rest at the top and bottom of its vibration, in very much the same manner as the reciprocating parts of an engine. Notice the varying forces shown by the spring balance. These are due to the varying accelerations of the vibrating weight. Instead of hanging the spring balance to a stand, hold it in your hand and let another student set the weight into vibration. The varying forces may now be felt and will give a clear insight into the matter.

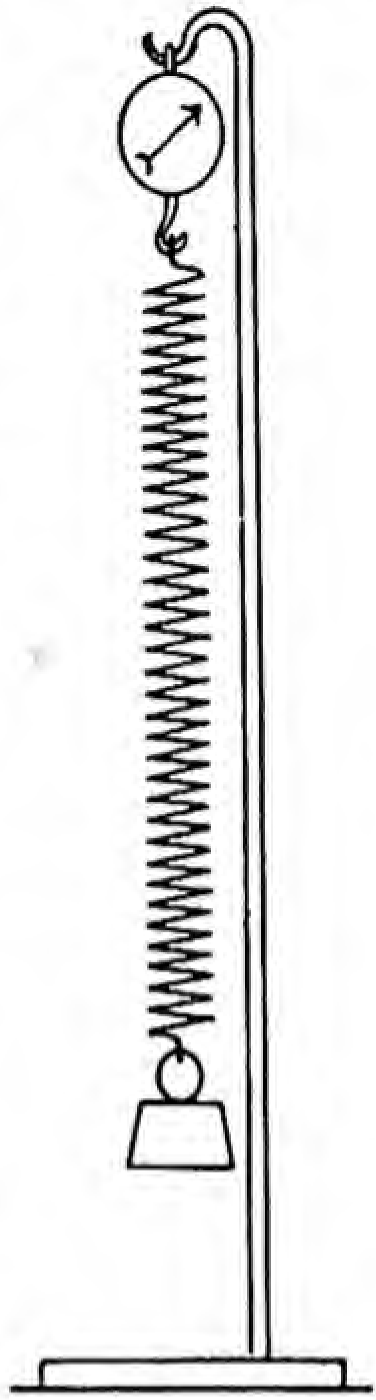


FIG. 114.—An experiment on inertia.

The calculation of the actual accelerations of the reciprocating parts is beyond the scope of this book. The following simple method of allowing for the forces required may however be employed.

When the crank is at X (Fig. 113), the crosshead will be at e, and the force then required to produce the necessary acceleration in the reciprocating parts will be given by

$$F_1=\frac{Wv^2}{gr}\left(1+\frac{r}{l}\right) \text{ lbs.} \quad\quad (1)$$

acting towards C.

When the crank is at Y, the crosshead will be at f, and the force now required will be

$$F_2=\frac{Wv^2}{gr}\left(1-\frac{r}{l}\right) \text{ lbs.} \quad\quad (2)$$

acting away from C.

In equations (1) and (2)

W=total weight of reciprocating parts, lbs.
v=velocity of crank pin, feet per second.
r=radius of crank, feet.
l=length of connecting rod, feet.
g=acceleration due to gravity.

To calculate the velocity of the crank pin,

Let N = revolutions of crank per minute.

Then $$\text{Revolutions per second} = \frac{N}{60},$$

$$v = 2\pi r \times \frac{N}{60} \text{ feet per second.}$$

It may be assumed that, when the crank and connecting rod are at right angles to each other (Fig. 115), the reciprocating parts

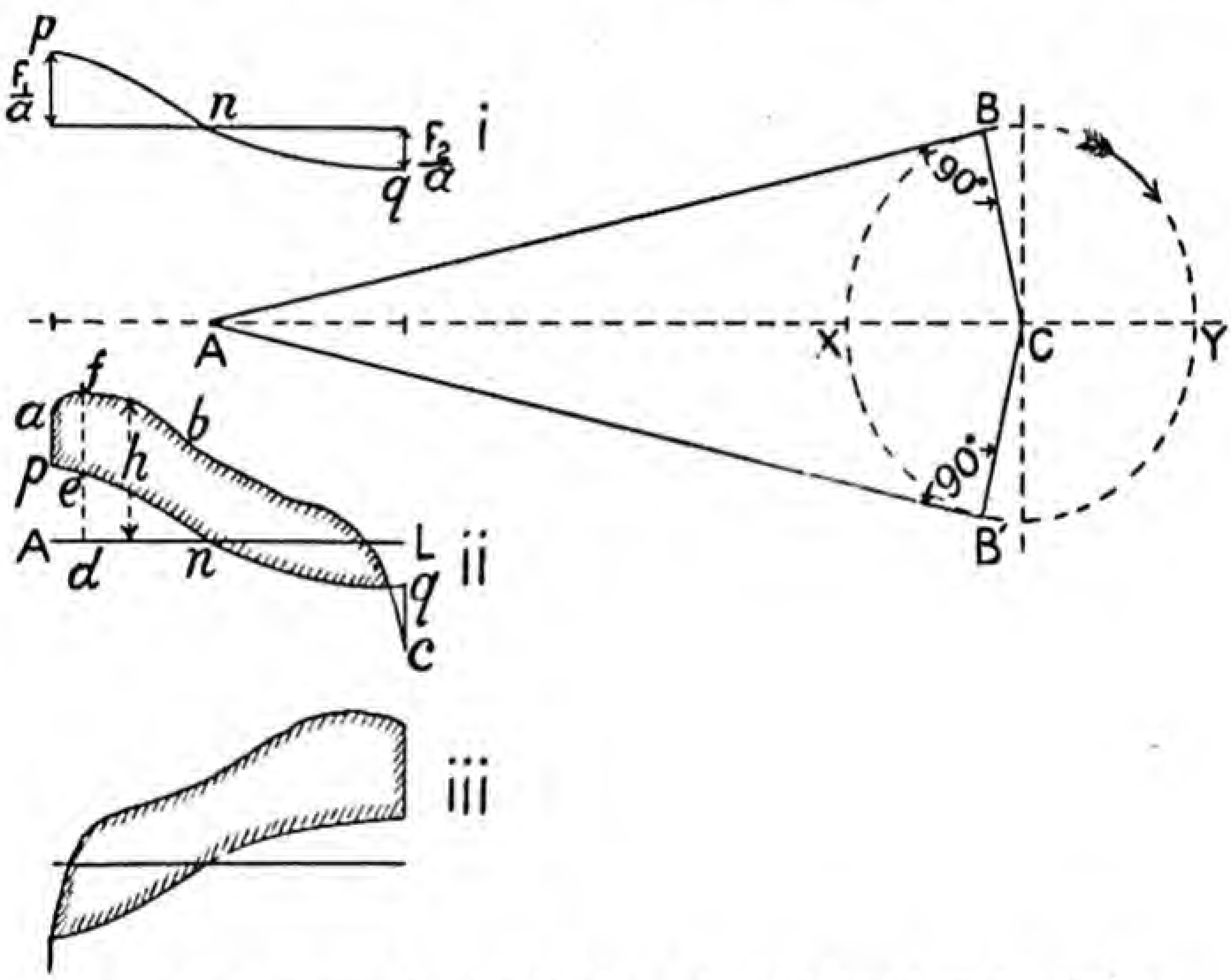


FIG. 115.—Diagrams showing the force given by the piston rod to the crosshead, with allowance for inertia.

have no acceleration and consequently no force is required in these positions. This will occur twice each revolution, as shown in Fig. 115 at B and B'. A diagram showing the force required to overcome **inertia per square inch of piston area** may now be drawn by taking a datum line projected from the ends of the stroke of the crosshead (Fig. 115 i), setting off an ordinate $\frac{F_1}{a}$, where a is the area of the piston in square inches, another ordinate $\frac{F_2}{a}$ below

the datum line to show that the force acts in the contrary direction, and finding the point n where there is no acceleration as above described. A fair curve drawn through pnq will give a good approximation and will enable the force required to overcome inertia for any piston position to be found readily. The scale of force most conveniently taken is one the same as that of the indicator diagram in Fig. 113.

Diagram with allowance for inertia.—The diagrams in Figs. 113 and 115 i may be conveniently combined so as to show the net resultant force with which the piston rod acts on the crosshead. Thus, taking the diagram for the left-to-right stroke as shown in Fig. 113 ii and reconstructing it as shown in Fig. 115 ii by taking a datum line AL and setting off ordinates h taken from the breadths gh in Fig. 113 ii, a curve abc will be produced which shows the net resultant pressures on the piston per square inch of its area, measured from AL. Now copy the diagram of inertia forces from Fig. 115 i giving the curve pnq. Any ordinate such as def being taken, df represents the net force on the piston, de represents the force required to overcome inertia, consequently the difference ef will give the net force per square inch of piston area which reaches the crosshead pin. Heights therefore between the shaded curves in Fig. 115 ii will give the desired information for any piston position. The return stroke diagram is shown in Fig. 115 iii.

It will be noticed that the general effect of inertia is to produce a much more equal effect on the crosshead than would be the case if the actual forces acting on the piston were to reach the crosshead.

Effect of inertia of connecting rod.—The inertia of the connecting rod modifies considerably the force which would otherwise be exerted on the crank pin. The connecting rod has two movements, one of **translation** in the direction of the line of the stroke and one of **rotation** about the centre of the crosshead pin. Both these movements undergo acceleration and consequently forces are required to act on the rod in order to overcome its inertia. The exact solution is much too complex to be described here. A simple approximation is to consider one portion of the weight of the rod to be reciprocating with the crosshead, this weight being added to the weight of the other reciprocating parts. The remainder of the weight of the rod may be considered as being

attached to the crank pin and simply rotating with it. In the case of a uniform rod the proportions would be half the weight to the crosshead and the other half to the crank pin.

Other effects of inertia.—The student will have observed in carrying out the experiment with the vibrating weight (p. 133) that forces were being communicated to the supports of the vibrating system, *i.e.* the stand or the body of the person holding the spring balance. Precisely the same effect occurs in the case of an engine. The inertia of all the parts produces reactions on the frame of the engine; such reactions being transmitted to the foundations may result in troublesome vibrations being set up in the surrounding buildings and may prove to be a

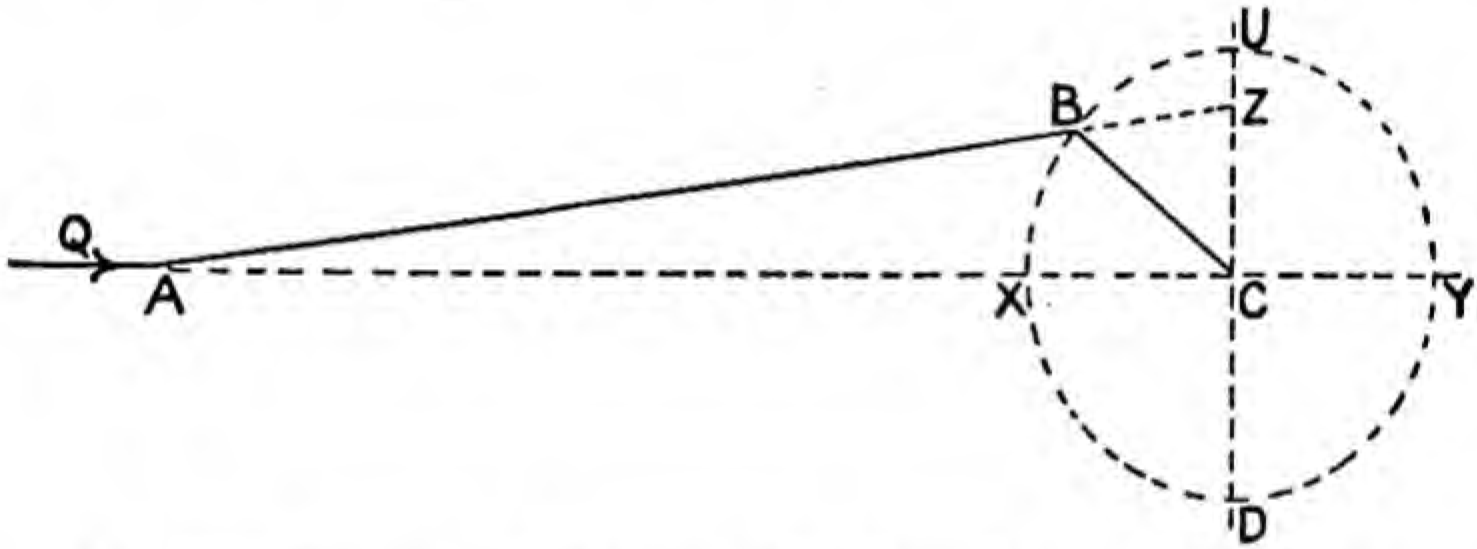


FIG. 116.—Construction for determining the turning moment on the crank.

nuisance. The effect is often particularly noticeable on board a ship driven by a propeller. In such cases, the engines are said to be out of balance. A perfectly balanced machine (for example a plain revolving disc) communicates no vibrations to its supports, that is to say, there is no resultant reaction given to its frame. The methods of securing balance are too complicated to be treated here; but the student will understand the principles to be followed, viz. there must be no resultant force or couple due to the inertia of the moving parts of the machine.

Turning moment diagram.—An important part of the work in the design of an engine consists in estimating the turning moment on the crank throughout the revolution in order that means may be taken for rendering it as uniform as possible. Let AB and BC be the connecting rod and crank (Fig. 116). Draw UD through C perpendicular to the line of stroke AC. Let Q lbs. be the net

resultant force given by the piston rod to the crosshead. Let Z be the point in which the connecting rod, produced if necessary, cuts the line UD. Measure CZ in feet to the scale of the diagram. Then it may be shown that

Turning moment

$$=T=Q\times CZ \text{ lb.-feet.}$$

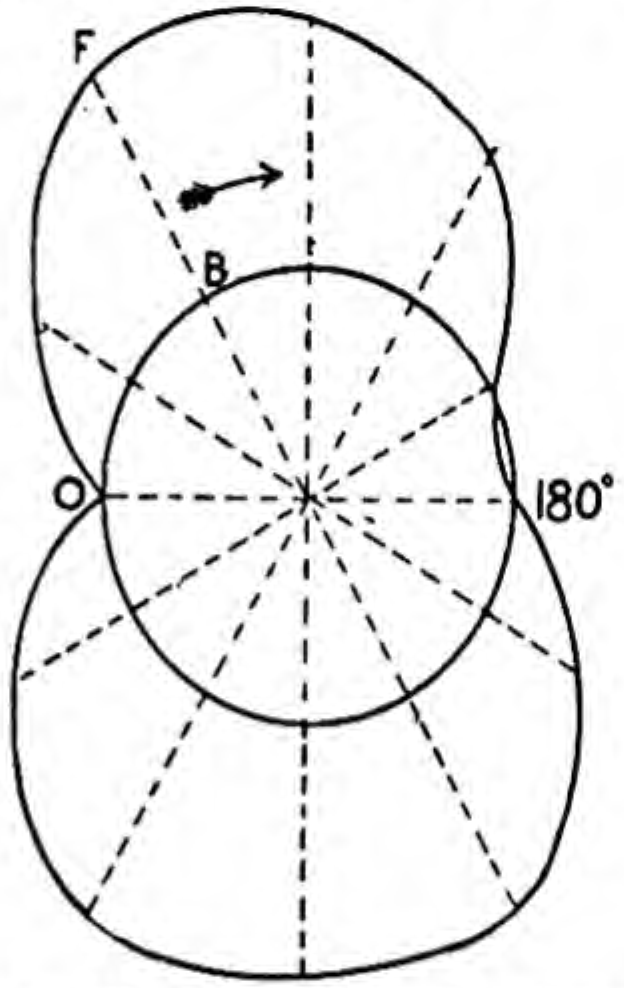


FIG. 117.—Turning moment diagram for a single crank engine.

To draw a diagram of turning moments, calculate T for crank angles differing by say 30° throughout the revolution and set off the calculated values along the direction of the crank outside the crank pin circle as shown at BF (Fig. 117). Draw a fair curve through the ends of the vectors, thus producing a **polar turning moment diagram.** The example shown in Fig. 117 is for a single crank engine; it will be noticed that the turning moment is zero when the crank pin is

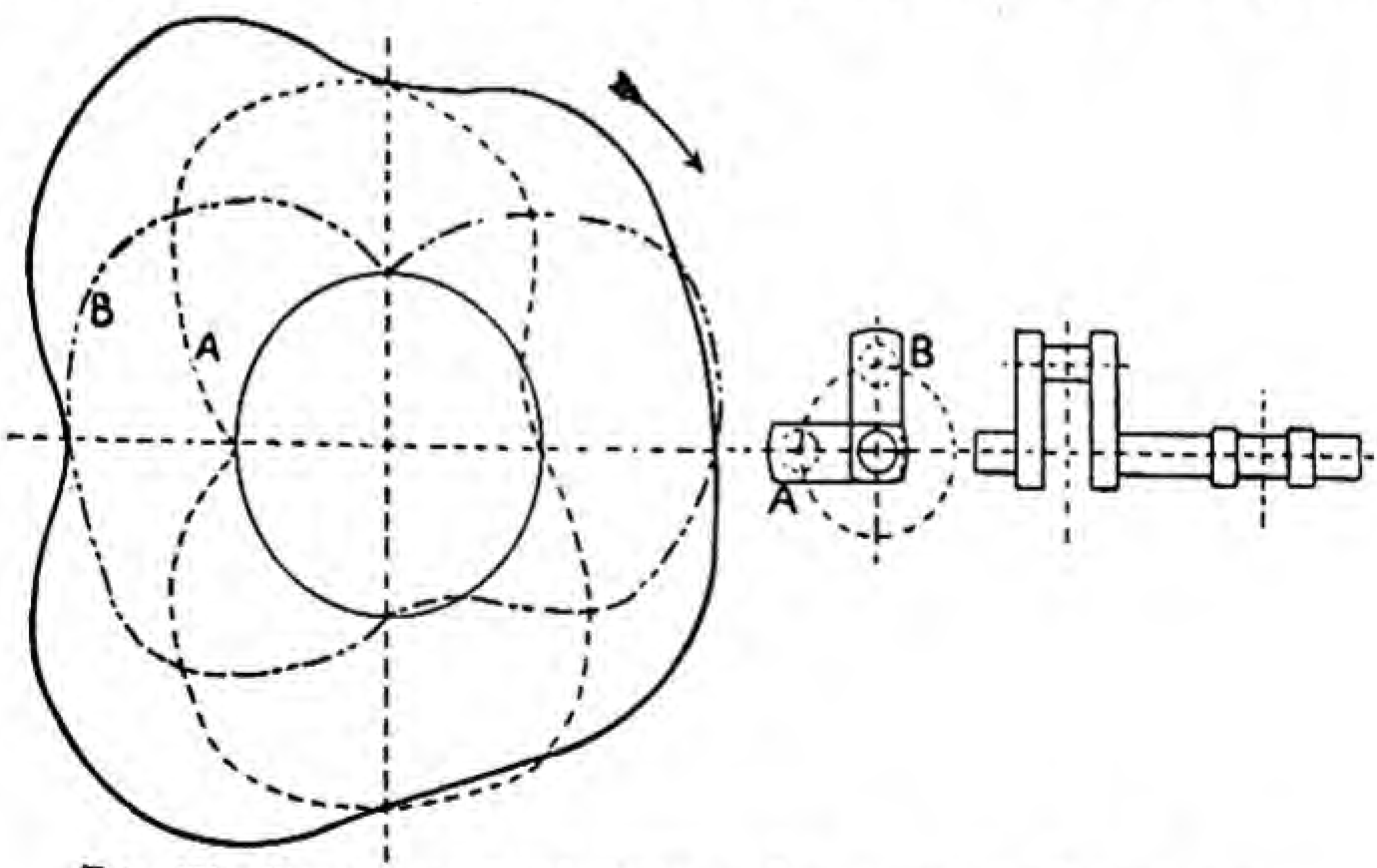


FIG. 118.—Turning moment diagram for an engine having two cranks at 90°.

passing through X and Y (Fig. 116)—hence the name given to these points, viz. **dead centres.**

In the case of engines having two or more pistons working on cranks at various angles, a turning moment diagram is drawn for each crank and a combined diagram is then drawn by adding together corresponding vectors. Fig. 118 shows such a diagram

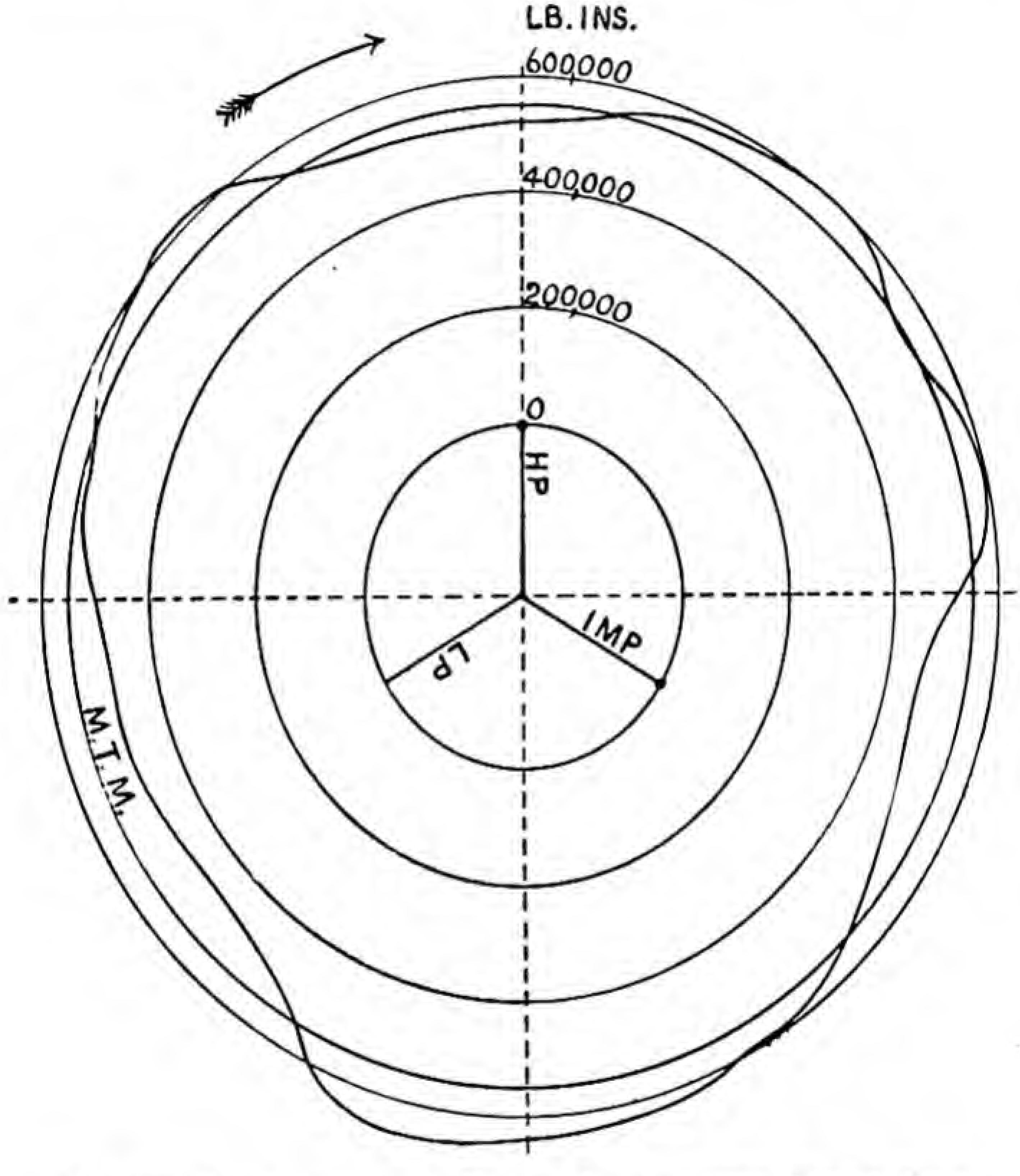


FIG. 119.—Turning moment diagram for an engine having three cranks at 120°.

for a two crank engine, cranks at 90°, and Fig. 119 gives a diagram for a three crank engine with cranks at 120°. The last diagram shows the uniformity which may be obtained by suitably arranging the cylinders and cranks; the engine for which it has been drawn is illustrated in Fig. 202.

Mean turning moment.—The mean turning moment acting on the crank shaft may be calculated from given particulars. Thus,

Let, T_m = mean turning moment, lb.-feet,

H.P. = horse-power developed on the cranks,

N = revolutions per minute.

Then, Work done in one rev. $= T_m \times 2\pi$ lb.-feet.

Work done per min. $= 2\pi T_m N$ lb.-feet.

$$\text{H.P.} = \frac{2\pi T_m N}{33{,}000}.$$

$$\therefore \quad T_m = \frac{\text{H.P.} \times 33{,}000}{2\pi N} \text{ lb.-feet.}$$

EXAMPLE. The total H.P. of the triple expansion engine having the turning moment diagram as shown in Fig. 119 is 1262 at 143 revolutions per minute. Calculate the mean turning moment.

$$\begin{aligned} T_m &= \frac{\text{H.P.} \times 33{,}000}{2\pi N} \\ &= \frac{1262 \times 33{,}000 \times 7}{2 \times 22 \times 143} \\ &= 46{,}330 \text{ lb.-feet.} \\ &= 556{,}000 \text{ lb. inches.} \end{aligned}$$

A circle MTM representing this mean turning moment has been drawn in Fig. 119 and enables the actual turning moment to be compared with the mean. The ratio of maximum turning moment to mean in this example is 1·187 to 1 ; and of minimum to mean 0·85 to 1.

Fluctuations in speed of rotation.—In general it is desirable that the speed of rotation of the engine shaft should be as uniform as possible. In attempting to secure this, the engineer has to consider the effect produced by the want of uniformity in the turning moment during each revolution, and also, in some cases, considerable alterations of short duration which may occur in the resistance offered by the machinery being driven by the engine. Assuming that the resistance offered is uniform, it will be noticed that the energy abstracted from the engine shaft will be taken at a uniform rate throughout the revolution. Should the turning moment applied by the engine to the shaft not be also perfectly uniform, and just sufficient to overcome the moment of resistance, so that the energy being delivered to the shaft is equal to the

energy being abstracted from it throughout the revolution, the shaft will be subject to changes of speed, and will even have a jerky action should there be great want of uniformity in the turning moment such as would occur in a single cylinder engine.

Excess energy given to the shaft above what is required to satisfy the demand must remain in the engine as additional kinetic energy in the moving parts, and these can only store additional energy of this kind by moving at a faster rate. As the energy being developed falls again, or, if the demand for energy increases, the excess kinetic energy is abstracted from the moving parts and the speed accordingly will diminish.

The student may consider one or two typical examples of sudden changes in the demand for energy and the consequent effect on the speed. Consider a machine for punching holes in metal plates driven direct by a steam engine. So long as no metal is being actually punched, the punch rises and falls, doing no work, and the only demand for energy is that required to overcome the comparatively small frictional resistances of the moving parts. The demand will increase enormously during the short time that a hole is being punched, and this must be provided for suitably. Again, consider the case of a steamship driven by a screw propeller. Should the ship be running in quiet water the propeller will be always under water, and the resistance offered to its rotation will be nearly constant. As the turning moment is also very uniform in such engines, the speed of rotation will be nearly constant. Suppose the ship to be among waves so that she is pitching, *i.e.* her stem and stern rising and falling. Circumstances may arise in such a case that will cause the propeller to be lifted out of the water for a few seconds, with the result that the greater part of the resistance to rotation is removed. The engines will then **race**, *i.e.* greatly increase the speed of revolution, unless proper provision is made to guard against this occurring.

Another kind of speed fluctuation is due to the supply of energy to the piston being greater or less than that required to supply the total demand for energy, such want of equality extending over a considerable period of time and so producing a steady rise or fall in the speed of revolution. Unless provision is made against this occurring, the speed of the engine may rise to a dangerous amount or may fall to zero.

Means of securing steady speed.—In the case of ordinary stationary engines, steadiness in the motion is obtained by two means. A heavy **flywheel** is mounted on the crank shaft so as to rotate with it, the object being to give a large addition to the mass of the moving parts of the engine and so to enable a much larger quantity of kinetic energy to be stored. Comparatively small and rapid fluctuations in the demand for energy are met from the large store of kinetic energy in the flywheel, which consequently changes its speed to a very small extent only. Jerky action is also thus avoided, whether due to want of uniformity in the resistances or in the turning moment. The great mass of the flywheel prevents, by reason of its inertia, any rapid or considerable change in its speed occurring such as would be produced in the crank shaft without it. The second means is to have a **governor** of some sort driven by the engine, the function of this appliance being to adjust the admission of steam, gas, or oil, to the cylinder so as to secure a supply of energy to the cylinder just sufficient to meet the demand. Such governors are arranged to reduce the supply of working stuff to the cylinder, should the speed of the engine increase above the proper amount, and to increase the supply should the speed fall. By the use of governors, the speed is thus maintained at a given average amount, and the deviation from this prearranged speed is kept small.

Many engines have no flywheel such as described above. In locomotives the great mass of the locomotive itself when in motion prevents any jerky action and thus takes the place of a flywheel. Flywheels are not used on board ship unless in the case of small auxiliary engines. Marine engines driving propellers have two or more cranks placed so as to secure great uniformity in the turning moment, and this is found to be sufficient. Modern paddle engines are also generally arranged with two cylinders so disposed as to equalise the turning moment. Want of space prevents the use of large flywheels on board ship and another more serious consideration is the **gyrostatic effect** of a heavy revolving wheel. Such wheels run well without producing forces on the bearings provided they are always kept rotating in one plane or in planes parallel to a given plane. But if any attempt is made to change the inclination of the plane of rotation a great resistance is encountered. The student who wishes to realise what would be

the effect on the bearings of a shaft carrying a heavy revolving flywheel in a ship among waves should perform the following experiment.

Expt. 29.—Take a bicycle wheel out of its frame and hold it in a vertical plane by means of the axle pin, one hand on each side of the wheel. Another person should now revolve the wheel rapidly. Let the student holding the wheel now attempt to turn it over so that its plane is horizontal. The resistance encountered will give a very keen appreciation of the effect on the bearings and hull of a ship in the case of a large wheel.

Energy stored in flywheels.—It is shown in mechanics that a body of mass m lbs. moving with a velocity of v feet per second possesses kinetic energy of an amount given by

$$\text{Kinetic energy} = \frac{mv^2}{2g} \text{ foot-lbs.},$$

g being the acceleration due to gravity, which may be taken as 32·2 feet per sec. per sec. for all places in Britain. We may find approximately the kinetic energy of a flywheel by considering its mass to be concentrated at the mean radius of the rim. Let v ft. per sec. be the velocity of a point at this mean radius, and m the mass of the wheel in lbs.; then

$$\text{Kinetic energy} = \frac{mv^2}{2g} \text{ foot-lbs.}$$

Example. Suppose the mass of the rim of a flywheel is 10 tons and its mean radius is 6 ft. Find its kinetic energy when rotating 90 times per minute.

$$\text{Revolutions per sec.} = \frac{90}{60}.$$

Velocity of a point on the wheel at 6 ft. radius

$$\begin{aligned} &= \text{mean circumference} \times \text{revs. per sec.} \\ &= 2\pi \times 6 \times \frac{90}{60} \\ &= \frac{396}{7} \text{ ft. per sec.} \end{aligned}$$

$$\begin{aligned} \text{Kinetic energy} &= \frac{mv^2}{2g} \text{ foot-tons} \\ &= \frac{10 \times 396 \times 396}{64 \cdot 4 \times 7 \times 7} \\ &= \underline{496 \cdot 9} \text{ foot-tons.} \end{aligned}$$

"M" of a flywheel.—Since the kinetic energy possessed by a body of given mass depends on the square of its velocity only, we may state that the kinetic energy of a given flywheel is simply proportional to the square of the revolutions per minute. If, therefore, we know the kinetic energy of the wheel at any given speed, say one revolution per minute, the energy at any other speed may be easily calculated. It is customary to use the letter **M** to denote the kinetic energy of a flywheel at one revolution per minute.

Let K = kinetic energy of a wheel at N revs. per min.

Then, $$\mathbf{M} : K = 1 : N^2,$$

or, $$K = \mathbf{M}N^2.$$

Fluctuation of speed in flywheels.—We may easily calculate the fluctuation in the speed of a flywheel if we know the fluctuation in the demand for energy. Thus, suppose a wheel of mass m lbs., to have a mean radius r feet; then, if the velocity of a point at the mean radius is v_1 ft. per sec. at a given instant,

$$\text{Kinetic energy} = K_1 = \frac{mv_1^2}{2g} \text{ ft.-lbs.} \qquad (1)$$

Suppose now that there is a demand for W ft.-lbs. of energy which has to be met by the store of energy in the wheel. The wheel will slow down while supplying this. Call v_2 the velocity of a point at the mean radius when the change is complete; then

$$\text{Kinetic energy} = K_2 = \frac{mv_2^2}{2g} \text{ ft.-lbs.} \qquad (2)$$

The wheel loses kinetic energy $= K_1 - K_2$, and this has been transformed into W ft.-lbs. of mechanical work,

$$\therefore \quad W = K_1 - K_2$$

$$= \frac{mv_1^2}{2g} - \frac{mv_2^2}{2g}$$

$$= \frac{m}{2g}(v_1^2 - v_2^2),$$

or, $$v_1^2 - v_2^2 = \frac{2g}{m} \cdot W. \qquad (3)$$

EXAMPLE i. A wheel of mass 2000 lbs. at a mean radius of 3 feet has a speed of 180 revolutions per minute. Suppose 4000 ft.-lbs. to be abstracted from it and calculate its new speed.

$$v_1=\frac{180}{60}\times 2\pi r$$
$$=3\times 2\times\frac{22}{7}\times 3$$
$$=\frac{396}{7}=56\cdot 57 \text{ ft. per sec.},$$

and $$v_1^2=3200.$$

Now, $$v_1^2-v_2^2=\frac{2g}{m}W$$
$$3200-v_2^2=\frac{64\cdot 4}{2000}\times 4000$$
$$=128\cdot 8.$$
$$v_2^2=3200-128\cdot 8,$$
$$v_2=\sqrt{3071}$$
$$=55\cdot 4 \text{ ft. per sec.}$$

Let new speed $=N_2$ revolutions per minute.

$$N_2=\frac{55\cdot 4\times 60}{2\pi r}$$
$$=\frac{55\cdot 4\times 60\times 7}{2\times 22\times 3}$$
$$=\underline{176\cdot 3} \text{ revolutions per min.}$$

The wheel therefore loses 3·7 revolutions per minute while giving up 4000 ft.-lbs. of energy.

EXAMPLE ii. Calculate **M** of the wheel in Example i.

At one rev. per min., $v=\frac{1}{60}\times 2\pi r$

$$=\frac{\pi}{10} \text{ feet per sec.}$$
$$\mathbf{M}=\frac{mv^2}{2g}$$
$$=\frac{2000\times\pi^2}{2\times 32\cdot 2\times 10\times 10}$$
$$=\underline{3\cdot 068} \text{ ft.-lbs.}$$

EXAMPLE iii. Calculate the energy stored in the same wheel at 120 revolutions per minute.

$$\text{Kinetic energy}=\mathbf{M}N^2$$
$$=3\cdot 068\times 120\times 120$$
$$=\underline{44180} \text{ ft.-lbs.}$$

Centrifugal governor.—The action of these governors has already been explained in Chapter I, p. 11. Suppose a body to be secured to the end of a cord and whirled in a circle. It will be found that an outward force acts constantly on the hand; this force is called **centrifugal force**, and its amount may be calculated from $F=\frac{wv^2}{gr}$ lbs.

where

w = weight of body in lbs.

v = velocity in circular path, feet per sec.

g = acceleration due to gravity = 32·2 feet per sec. per sec.

r = radius of circle in feet.

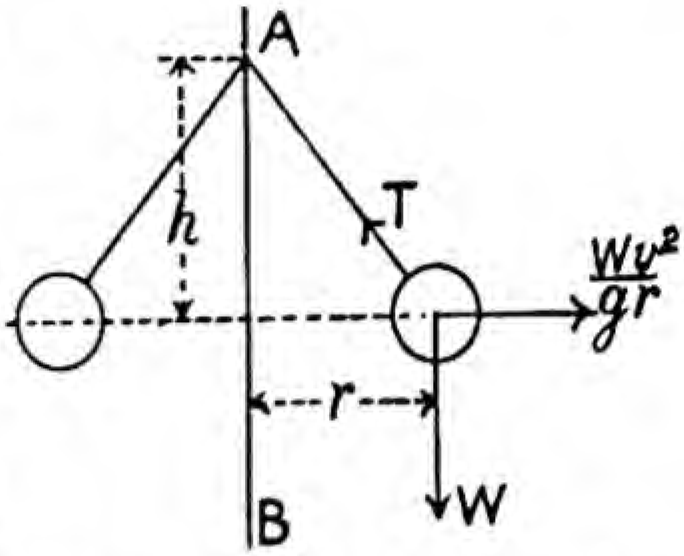


FIG. 120.—Diagram showing the forces acting on a ball of a simple governor.

Consider the forces acting on one of the balls of the simple governor shown in outline in Fig. 120. There will be its weight, W, the pull of the arm T and the centrifugal force $\frac{Wv^2}{gr}$. If the ball is rotating at a steady speed it will keep at a constant radius r from the axis AB, and these forces will be in equilibrium. Let h be the height in feet from the plane of revolution of the balls to A. Then it can easily be shown by an application of the parallelogram of forces that

$$\frac{Wv^2}{gr} : W = r : h.$$

It therefore follows that $\frac{Wv^2}{gr}h = Wr$,

$$h = g\frac{r^2}{v^2}. \qquad \text{(1)}$$

Let N = revolutions per minute of the ball, then

$$v = \frac{2\pi rN}{60},$$

giving by substitution in (1) $h = g\frac{r^2 \times 60 \times 60}{4\pi^2 r^2 N^2}$

$$= \frac{3600g}{4\pi^2} \times \frac{1}{N^2} \text{ feet}$$

$$= \text{a constant} \times \frac{1}{N^2} \qquad \text{(2)}$$

This result shows that h is independent of the weight of the ball and of the length of the arm, and depends only on the reciprocal of the square of the revolutions per minute in this type of governor. A governor of this class, however, is only suitable for low speeds of revolution, as at higher speeds the arm to which the ball is attached comes nearly to the horizontal. The arm works best at an angle of about 45°, as then a decided movement is given to the sleeve for a small change in speed. For higher speeds a loaded governor, such as is shown in Fig. 19, p. 12, is required.

In a governor of the last mentioned pattern, it may be shown that

$$h=\frac{900g}{\pi^2}\left(1+\frac{W_1}{W}\right)\frac{1}{N^2}\text{ feet}$$

$$=\text{a constant}\left(1+\frac{W_1}{W}\right)\frac{1}{N^2}.$$

Where W_1 = weight of the dead load,
W = weight of one ball,
N = revolutions per minute of spindle.

We therefore deduce from this equation that any increase given to W_1, will increase the value of h, thus bringing the balls nearer the spindle for the same speed of revolution. To make the balls rotate at the same radius as before, the speed of revolution must be increased.

Stability of governors.—For a governor to work properly it must possess **stability.** A stable governor has a definite position of balls for each speed, and any small change in the speed produces a correspondingly small change in h. The governor is unstable if a small change in speed produces a large change in h. The governor should be **sensitive**, *i.e.* respond promptly so as to produce the required change in the position of the throttle valve. Too great sensitiveness will cause the governor to overdo its work by opening the throttle valve more than is required to counteract a slight fall in speed. The result will be that the engine will increase its speed beyond the proper amount, when the over-sensitive governor will again close the throttle valve too much, thus producing too great a fall in speed. These fluctuations in speed going on continuously are called **hunting.** The engine goes hunting, as it were, above and below its proper speed.

EXERCISES ON CHAPTER IX.

1. Explain the meaning of the term "turning moment." An engine crank is 1 foot, and the connecting rod $4\frac{1}{2}$ feet, in length. Draw the crank and connecting rod in the position when the angle between them is 90°. In this position the force exerted by the piston rod is 4500 lbs. Find the turning moment.

2. In question 1 find the total pressure on the guide for the given position. What area of sliding surface must the slipper have if the pressure on it is not to exceed 70 lbs. per square inch?

3. Explain clearly the reason for often providing two slippers, one on each side of the crosshead, in engines which frequently reverse.

4. A steam engine cylinder is 20″ diameter. The steam pressure on one side of the piston at a certain instant is 65 lbs. per square inch and on the other side is 17 lbs. per square inch. Calculate the resultant force acting on the piston.

5. Answer question 4 assuming that the piston rod is $3\frac{1}{2}$″ diameter and is on the low pressure side of the piston.

6. Explain clearly the reasons why the resultant force on the piston as calculated in question 5 does not arrive at the crosshead with the calculated value.

7. The reciprocating parts of an engine weigh 400 lbs. At a given instant their acceleration is 250 feet per sec. per sec. Calculate the force required to overcome the inertia.

8. Using dimensions of crank and connecting rod as in question 1, calculate the accelerations of the reciprocating parts at the inner and outer dead points when running at 150 revolutions per minute.

9. In question 1, assume that the reciprocating parts have no acceleration in the given position and, using the results found in question 8, draw an approximate curve showing the force required to overcome inertia during the forward or out-stroke of the piston.

10. Explain in general terms the necessity for balancing engines.

11. An engine is developing 100 H.P. at 180 revolutions per minute. Calculate the mean turning moment.

12. Contrast the action of the flywheel and of the governor in controlling the speed of an engine.

13. A flywheel has a mean radius of 5 feet. Its mass is six tons and it runs at 120 revolutions per minute. Calculate its kinetic energy.

14. Suppose the flywheel in question 13 to give up 55,000 ft.-lbs. of energy, what will be its speed then?

15. Find the **M** of the wheel in question 13. Calculate what energy will be stored in the wheel at 90 revolutions per minute.

16. Explain the action of a simple Watt governor. Why are modern governors usually loaded?

17. In a loaded governor the balls each weigh 3 lbs., the dead load weighs 20 lbs., and the balls rotate 240 times per minute. Calculate h in inches.

18. Calculate the value of h in question 17, supposing that the speed of the governor is diminished 2 per cent.

19. Describe with sketches how any governor keeps the speed of an engine fairly constant. What is meant by *hunting?* 1904.

20. Explain why both the flywheel and governor are needed to regulate or govern the speed of an engine. 1905.

21. If a piston with its rod weighs 250 lbs., and if at a certain instant when the resultant total force due to steam préssures is 3 tons the piston has an acceleration of 320 feet per second per second in the same direction, what is the actual force acting on the crosshead? 1905.

22. F lb. is the outward radial force on each ball of a governor required to keep it in equilibrium at the distance r feet from the axis when not revolving. The following are for extreme cases:

r	F
0·5	100·1
0·7	144·6

The weight of each ball being 10 lbs., what is the centrifugal force of each at n revolutions per minute, the radius being r? What are the speeds for the above values of r when the governor is revolving? 1906.

23. Why is an engine *balanced?* Describe generally any method of balancing the rotating parts that is known to you.

Imagine a long railway truck containing an invisible caged lion on a level track; axle bearings frictionless; imagine the lion to walk backwards and forwards to the limits of its cage, what would an outsider observe? Now suppose the wheels of the waggon blocked, what occurs? 1903.

CHAPTER X.

BOILERS.

Boilers.—The function of the boiler is to furnish a supply of steam at the required pressure, of quality as nearly dry as possible. There are many different types of boilers, and in the selection of one to work under given conditions, attention must be paid to

(*a*) The suitability of the design for safely carrying the proposed steam pressure.

(*b*) The efficiency with which the potential energy of the coal or other fuel is converted into heat in the steam.

(*c*) Accessibility, both externally and internally.

(*d*) Space occupied.

(*e*) Speed with which steam can be raised, starting with cold water and fires out.

Boilers are either **fire-tube** or **water-tube.** In fire-tube boilers, the furnace gases are led through tubes around which water circulates. In water-tube boilers, the water occupies the interior of the tubes and the gases pass over them externally. In general, water-tube boilers are more suitable than fire-tube boilers for the generation of steam at very high pressures, and for the fulfilment of condition (*e*) above. Fire-tube boilers are suitable for steady working at pressures up to say 200 lbs. per square inch, and can be constructed so as to fulfil condition (*c*) above very completely. The efficiency in well-designed boilers of both types is about the same. Water-tube boilers take up less space than fire-tube boilers of the same power.

Only a few of the best known kinds of boilers can be described here. Although water-tube boilers are extensively used for land

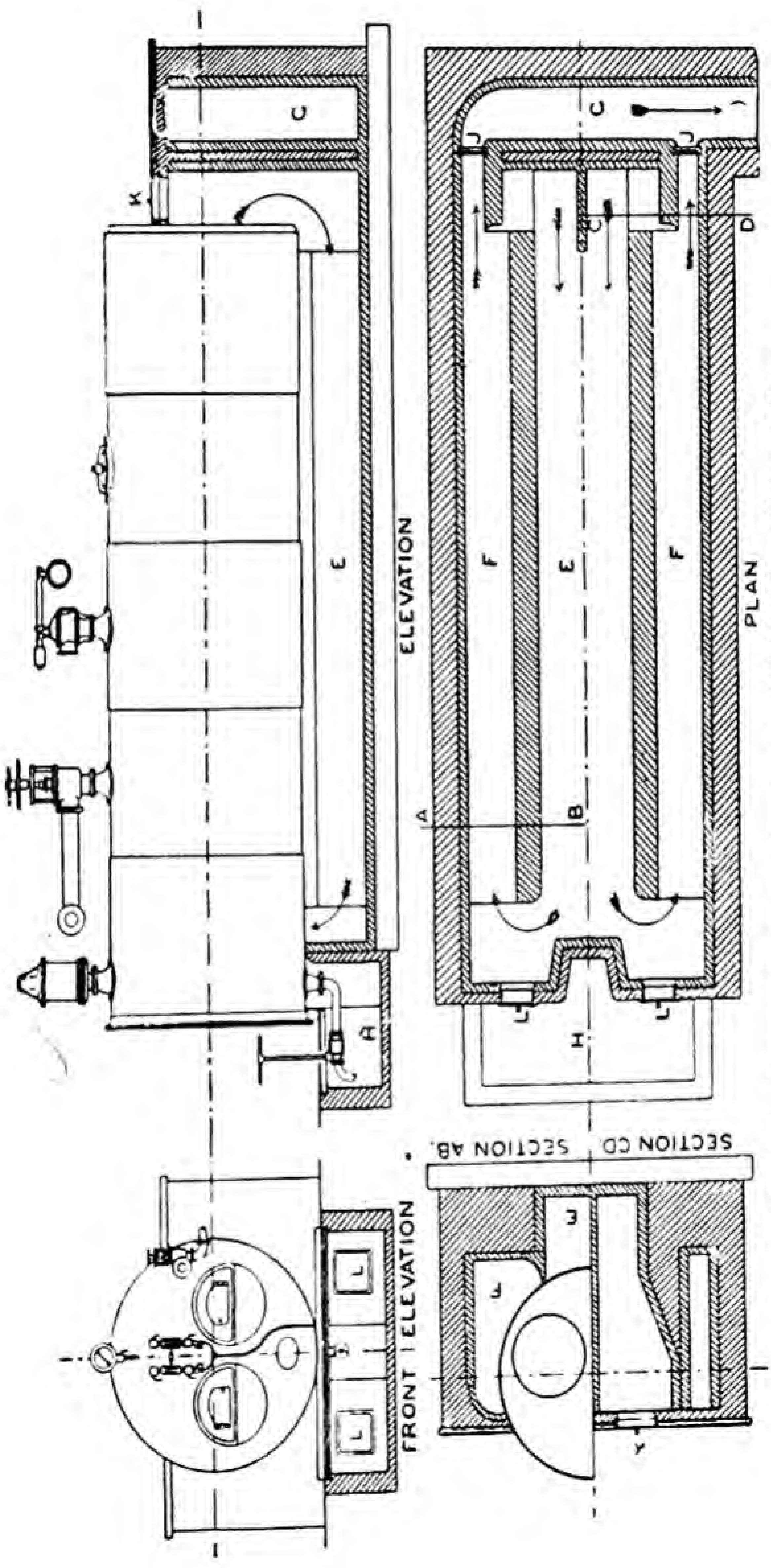


FIG. 121.—Lancashire boiler, showing the brickwork seating and flues.

purposes, yet, for general factory purposes, the Lancashire boiler is the most popular.

The Lancashire boiler.—The construction and arrangement in this boiler will be understood by reference to Fig. 121, in which is shown a complete boiler with its brickwork seating and flues, to Figs. 122 and 123, showing the finished shell, and to the other illustrations in which the shell details are given. The drawings

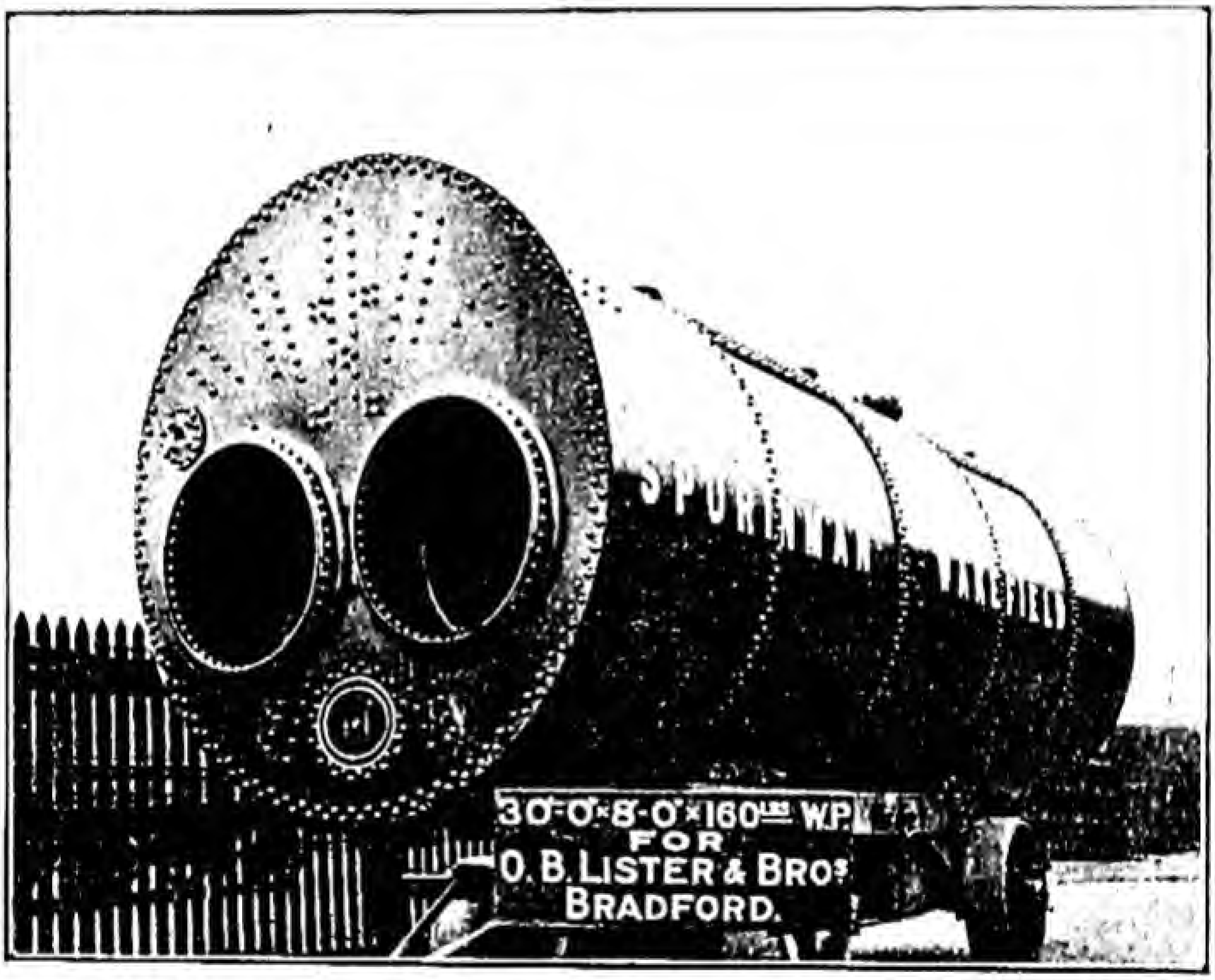


FIG. 122.—Finished shell of a Lancashire boiler.

are reproduced here by the courtesy of Messrs. Spurr, Inman & Co., Ltd. Referring to Fig. 122 it will be observed that the boiler consists of a large cylindrical shell, generally from 25 to 30 feet long and from $6\frac{1}{2}$ to 9 feet in diameter. Two large tubes of diameter about 0·4 that of the shell pass from end to end. A furnace, *C*, about 6 feet long, is placed at the front end of each tube (Fig. 123). The hot gases from the burning fuel pass along the furnace tubes, emerging at the back end, where they pass downwards and unite in a bottom flue *E* (Fig. 121). Passing along *E*, the gases divide at the front end of the boiler into side

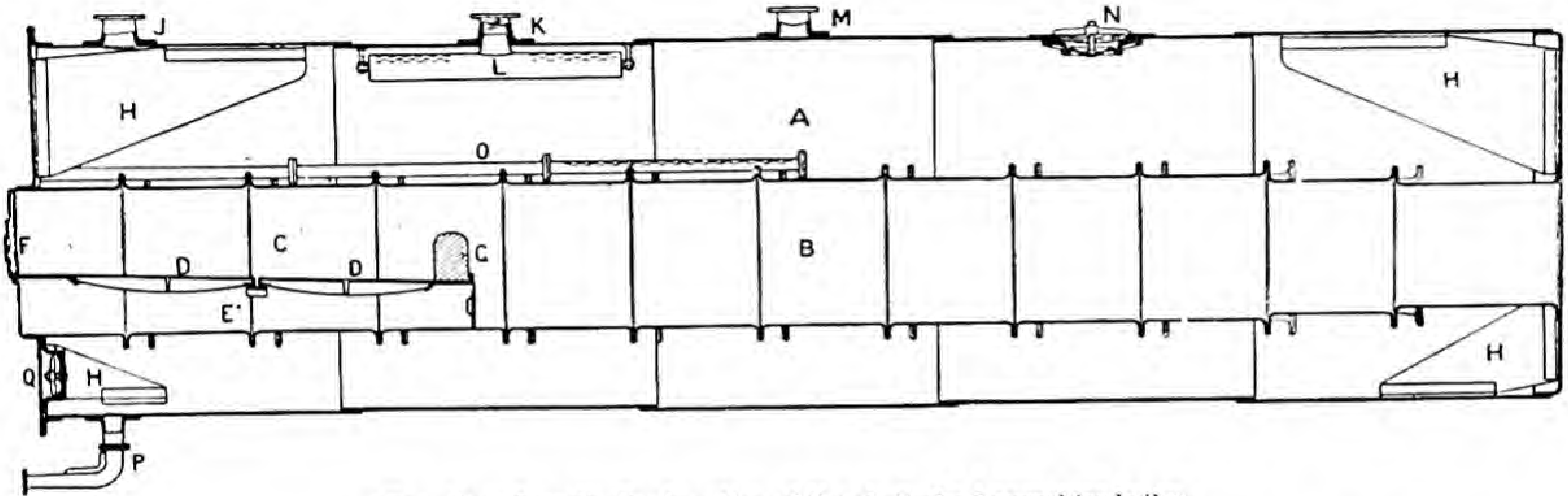


Fig. 123.—Longitudinal section of the shell of a Lancashire boiler.

A. Shell.
B. Furnace tube.
C. Furnace.
D. Fire bars.
E. Ash pit.
F. Furnace door.
G. Fire bridge.
H. Gusset stays.
J. Mounting block for dead weight safety valve.
K. Mounting block for junction valve.
L. Anti-priming pipe.
M. Mounting block for low water and high steam safety valve.
N. Manhole door.
O. Internal feed pipe.
P. Blow-off pipe.
Q. Mud hole door.

flues *FF*, and again travel to the rear of the boiler, where they make their exit to the chimney through the flue *G*. The flues are built of common brick, lined with firebrick in order to withstand the heat. **Dampers** consisting of doors sliding vertically in frames are placed at *JJ*; these serve to regulate the draught. Doors by means of which access can be obtained to the flues are placed at *K*, *L*, *L*. *H* is a shallow pit, called the **blow-off pit**, covered with iron plates forming part of the floor of the boiler room. The

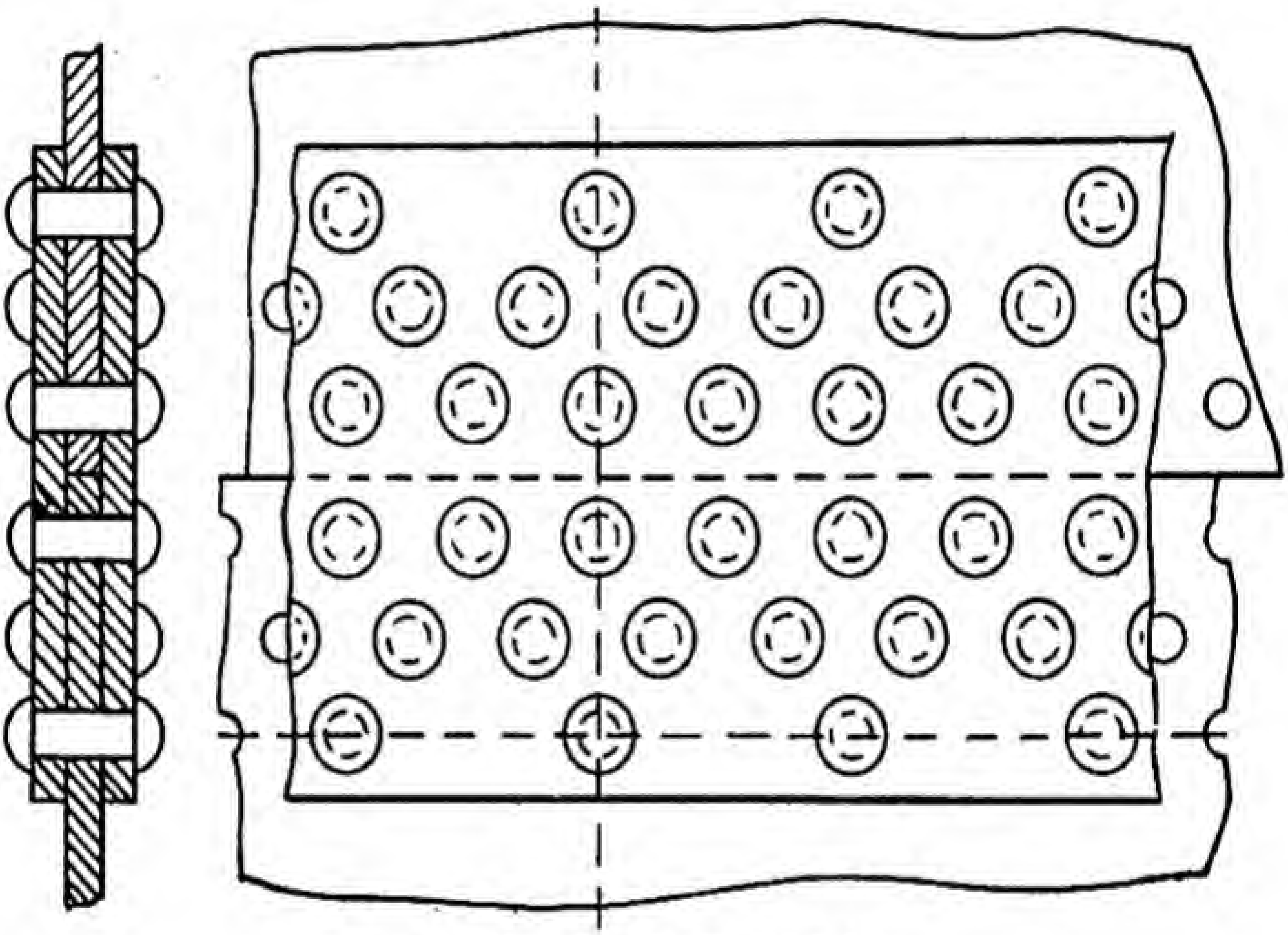


FIG. 124.—Longitudinal joint.

valves and other mountings shown in this illustration will be described in detail in Chap. XI.

Construction of shell.—In the boiler shown in Figs. 122 and 123 there are five rings in the cylindrical shell, each ring being constructed from a single plate bent into a truly cylindrical form. The meeting edges of the plate are connected by a butt joint with cover plates inside and outside and six rows of rivets (Fig. 124). These joints form the **longitudinal joints** of the shell. The alternate rings of the shell are made smaller in diameter than the others by an amount equal to twice the thickness of the plates. This permits the rings to be secured together by lap joints, usually of the double riveted form (Fig. 125); these are the **circumferential joints.**

The longitudinal joints are made much stronger than the circumferential joints because the stress on them due to the internal pressure is double that on the circumferential joints (p. 207). In order to avoid a continuous row of rivets, the longitudinal joints are arranged to be out of line with one another in the successive rings of the shell, and are placed near the top of the boiler above the brickwork seating where they will be accessible; this arrangement also removes these joints from contact with the hot flue gases, the action of which might produce deterioration.

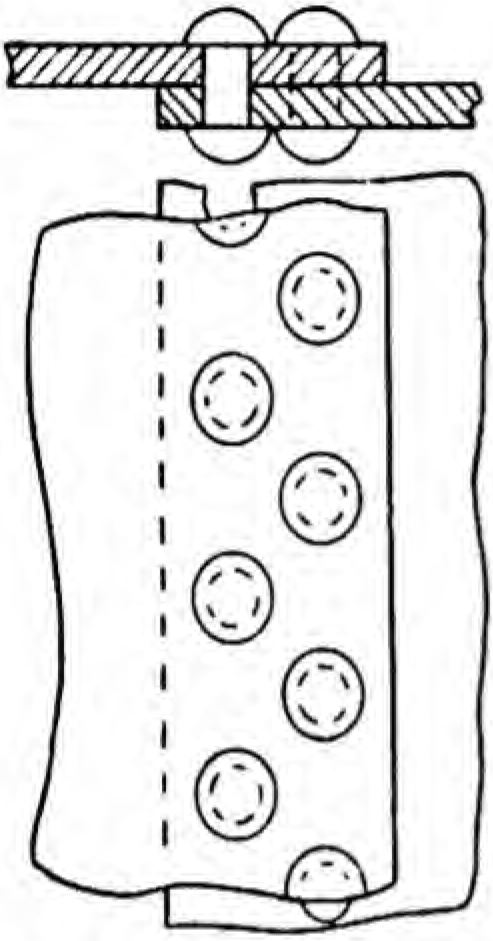


FIG. 125.—Circumferential joint.

Furnace tubes.—Furnace tubes are constructed of rings bent to a truly cylindrical form, the meeting edges being welded together. The ends of each ring are flanged outwards (Fig. 127) and the rings are connected together, end to end, by rivets passing through the flanges, a welded expansion ring being placed between the flanges in order to facilitate **caulking**, by means of which a steam-tight joint

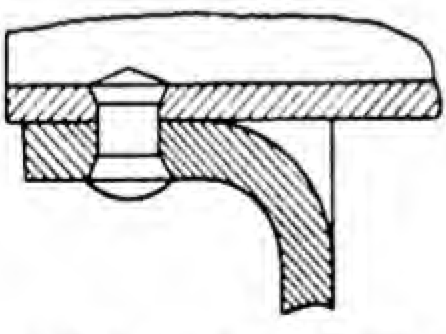


FIG. 126.—Attachment of furnace tube to front end plate.

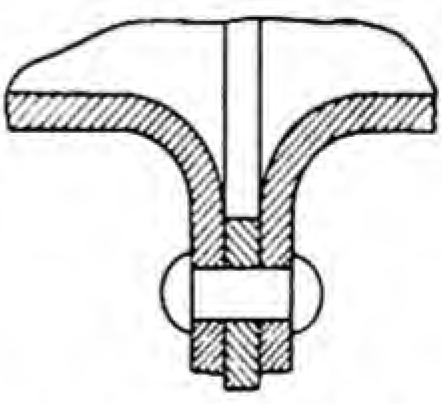


FIG. 127.—Adamson's flanged ring for permitting the furnace tubes to expand.

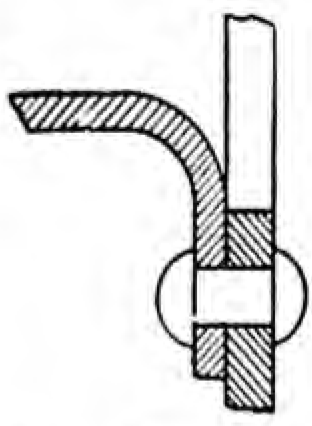


FIG. 128.—Attachment of furnace tube to back end plate.

is secured. Notice that in this arrangement the heads of the rivets are in the water space of the boiler, where they will not be exposed to the direct action of the fire. The joint is known as the **Adamson flanged ring**, and gives a considerable amount of elasticity to the tube in the direction of its axis. This elasticity permits of

the expansion of the furnace tubes in the direction of their length, due to their being at a higher temperature than the shell, being taken up without severely stressing any part of the boiler. Furnace tubes are exposed to external pressure tending to collapse them. To strengthen them against this pressure, the rings are made of short length in order to have the supporting effect of many Adamson's flanged rings. The front end plate of the boiler has holes cut in it to receive the tubes; the edges of the holes are flanged outwards and the tube is slipped inside and secured by riveting (Fig. 126). The connection to the back end plate is made by the flange on the last ring of the tube (Fig. 128). The back

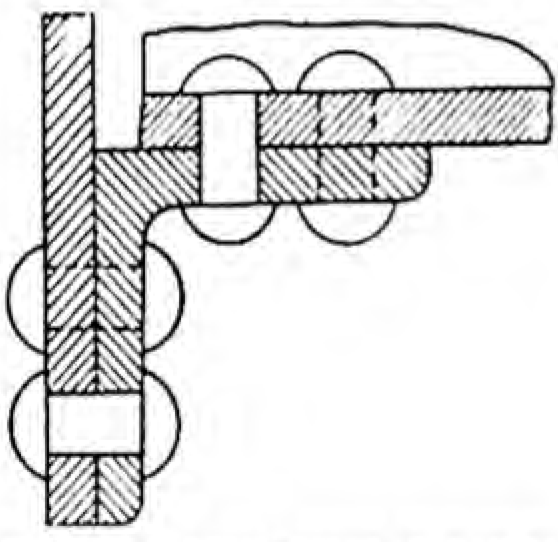


FIG. 129.—Attachment of shell to front end plate.

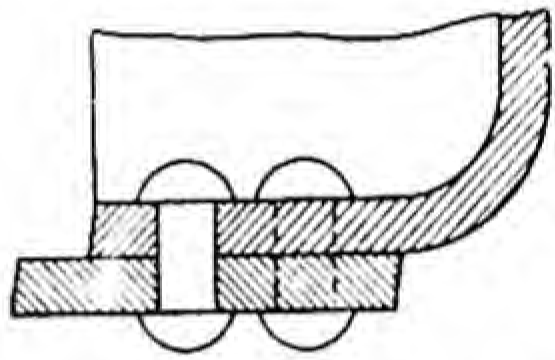


FIG. 130.—Attachment of shell to back end plate.

end rings of the furnace tubes, it will be noticed, are made smaller in diameter (Fig. 123). This plan provides a greater space between them, and gives access to the bottom of the boiler under the tubes for cleaning and examination.

End plates.—Each end plate is constructed from a single plate. The front end plate is connected to the shell by means of an outside angle hoop, bent to shape and having its meeting ends welded together to form a solid ring. Connection of this ring to the end plate and also to the shell is made by a double row of rivets (Fig. 129). The back end plate is flanged into the shell and double riveted to it (Fig. 130); an angle hoop joint is inadmissible at the back end as the angle would be exposed to the direct action of the furnace gases, and would be liable to burning. The angle connection of the front end plate, it should be remarked, permits

rather more freedom to this end to bulge than the flanged joint at the back will allow to the back end. No flat part of a boiler can

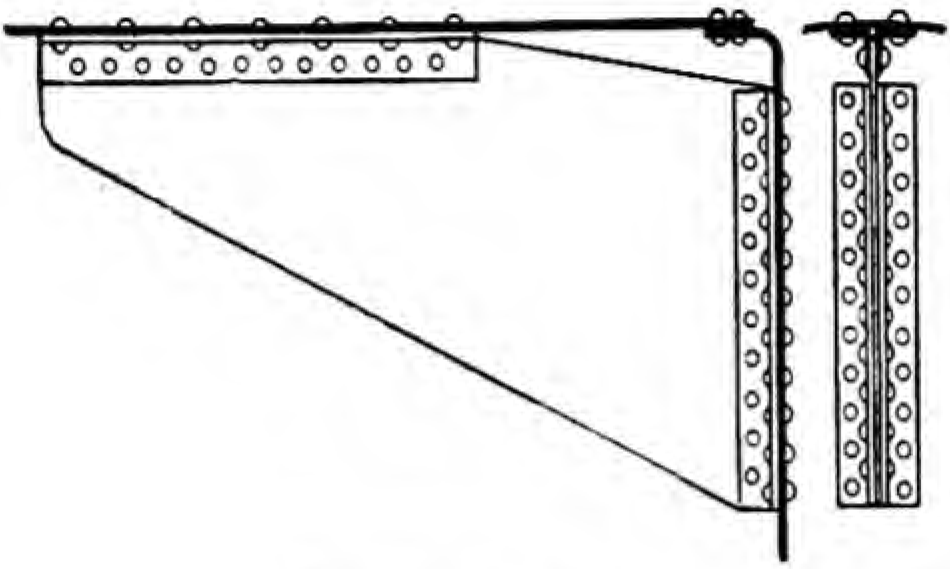


FIG. 131.—Gusset stay.

retain its shape without bulging unless it is supported or **stayed**. In the boiler under consideration, the ends are stayed to the

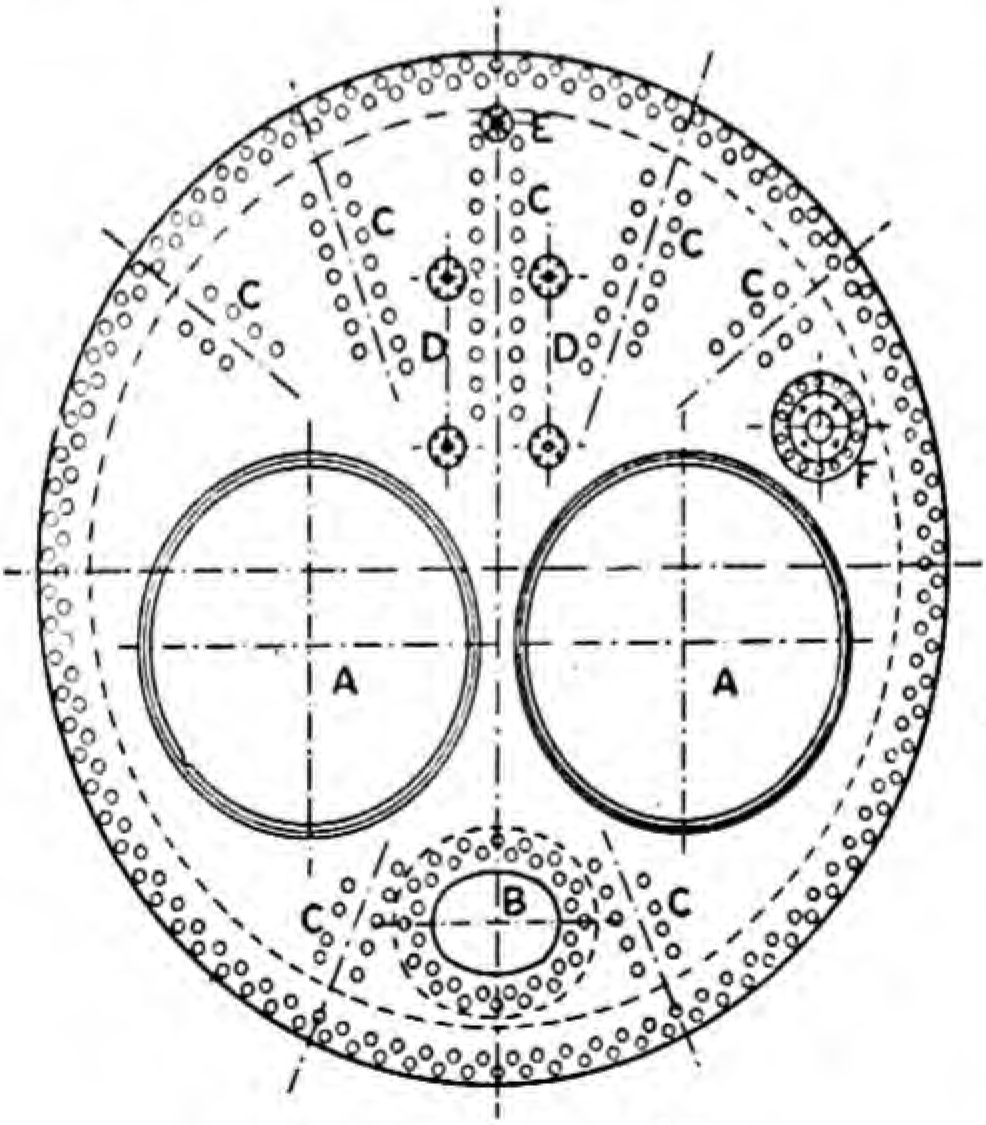


FIG. 132.—Front end plate.

A. Furnace tubes. **B.** Mud hole. **C.** Gusset stays. **D.** For water gauges. **E.** For pressure gauge. **F.** For feed check valve.

cylindrical shell by means of **gusset stays** (Fig. 131). Some of these are placed above the furnace tubes and some below, as will

be seen by inspection of Figs. 132 and 133, in which the front and back end plates are shown. The gusset stays must not come too close to the joints of the tubes with the end plates, otherwise the ends will be too stiff. A small amount of freedom to bulge is allowed round the furnace tubes in order to accommodate the expansion of the tubes on heating. The tops of the furnace tubes are usually at a higher temperature than the bottoms, therefore the expansion will be greater at the top than at the bottom. This

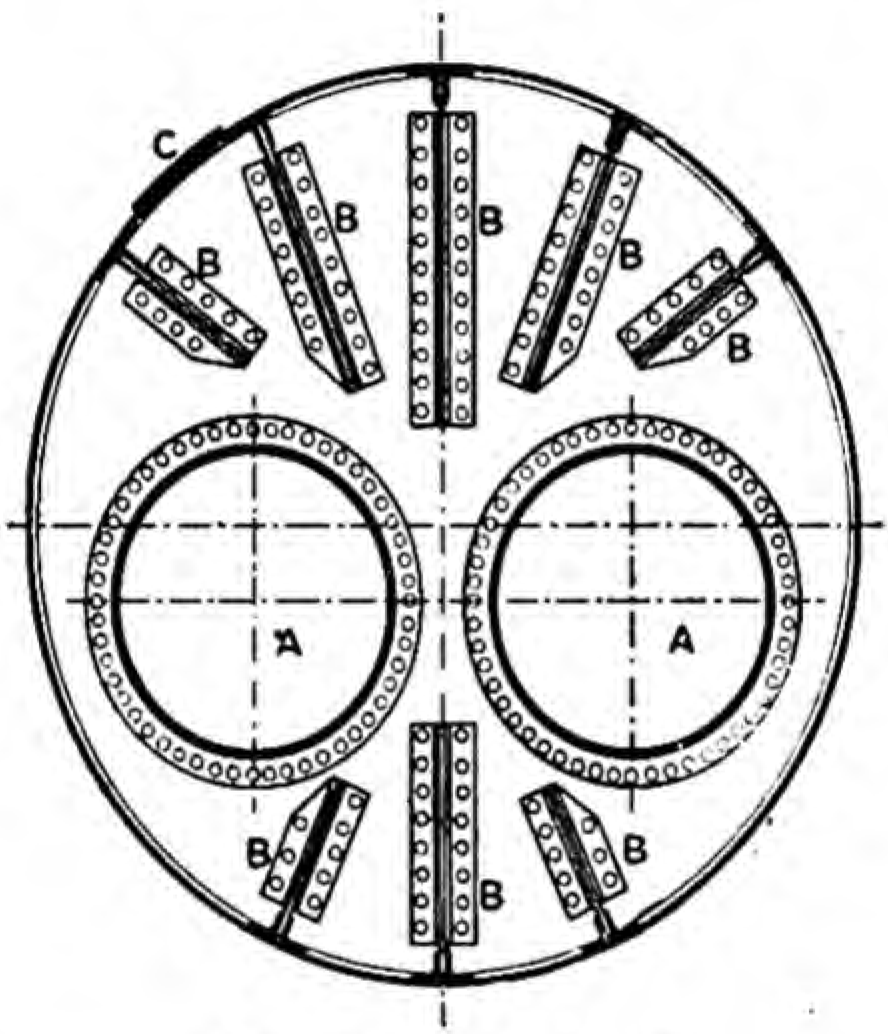


FIG. 133.—Back end plate.

A. Furnace tubes. B. Gusset stays. C. Longitudinal joint.

causes the tubes to rise at the middle of their lengths, producing **hogging.** Each time the furnace door is opened, the rush of cold air cools the tube, which consequently contracts. Action such as this, occurring say every half-hour, will ultimately, if the end plates are too stiff, cause a groove to be formed in the metal of the tube near the top on the water side, just where the tube is connected to the front end plate.

Mounting blocks.—Mounting blocks for receiving the various mountings, such as safety valves, steam valve, etc., are usually made of mild or cast steel, and are riveted strongly to the shell. The object of these blocks is twofold: they provide a faced joint to

receive the flange of the valve, etc., and also they strengthen the shell which had previously been weakened by the removal of the material in forming the hole. Mounting blocks are sometimes in the form of short conical pipes flanged at both ends, and these serve for the larger mountings; others consist merely of flat strength-compensating rings riveted to the boiler plates, such being used for the smaller mountings like water-gauge cocks.

Furnaces.—The furnaces consist of a fire grate constructed of fire bars placed side by side with an air space between each pair. The fire bars are supported on bearers resting on brackets riveted to the sides of the furnace tubes. Generally, the furnace

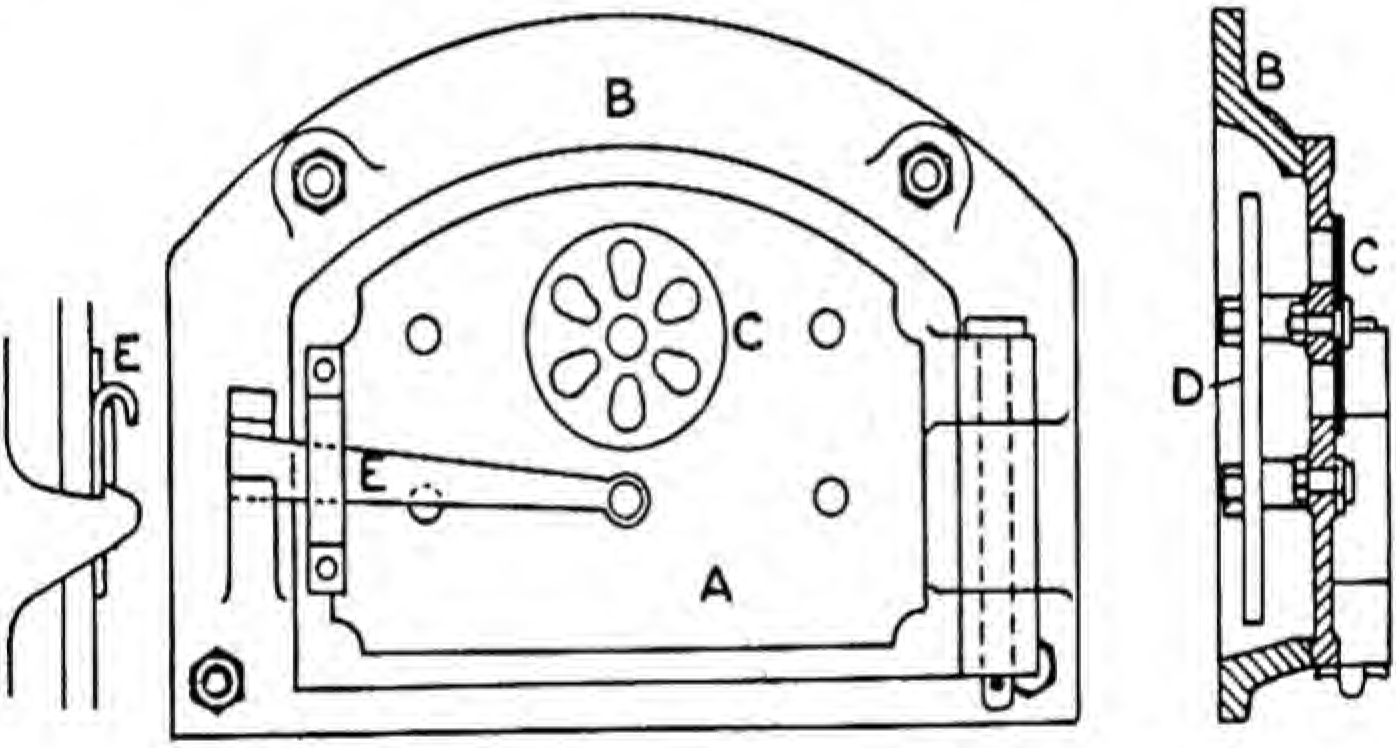


FIG. 134.—Front elevation and section of a furnace door.

is long enough to necessitate two lengths of fire bars being used. The furnace terminates at the back end in a firebrick **bridge**, the objects of which are to retain the fuel from falling over the end of the furnace, and also to restrict the space through which the gases coming from the fire have to pass. The latter is necessary to produce proper mixture of the air and the gases, and to give perfect combustion. Details of a furnace may be studied in Fig. 145. The furnaces are closed at the front ends by furnace doors *A* (Fig. 134), hinged to door frames *B*, and provided with adjustable openings at *C*, through which air may enter the furnace above the grate bars. Air is usually required just after fresh fuel has been fed to the furnace in order to prevent the formation of black smoke. The door is secured by

a hasp *E*. and has a cast-iron guard plate *D* secured on the inner side by distance pieces, so as to prevent the heat from the furnace

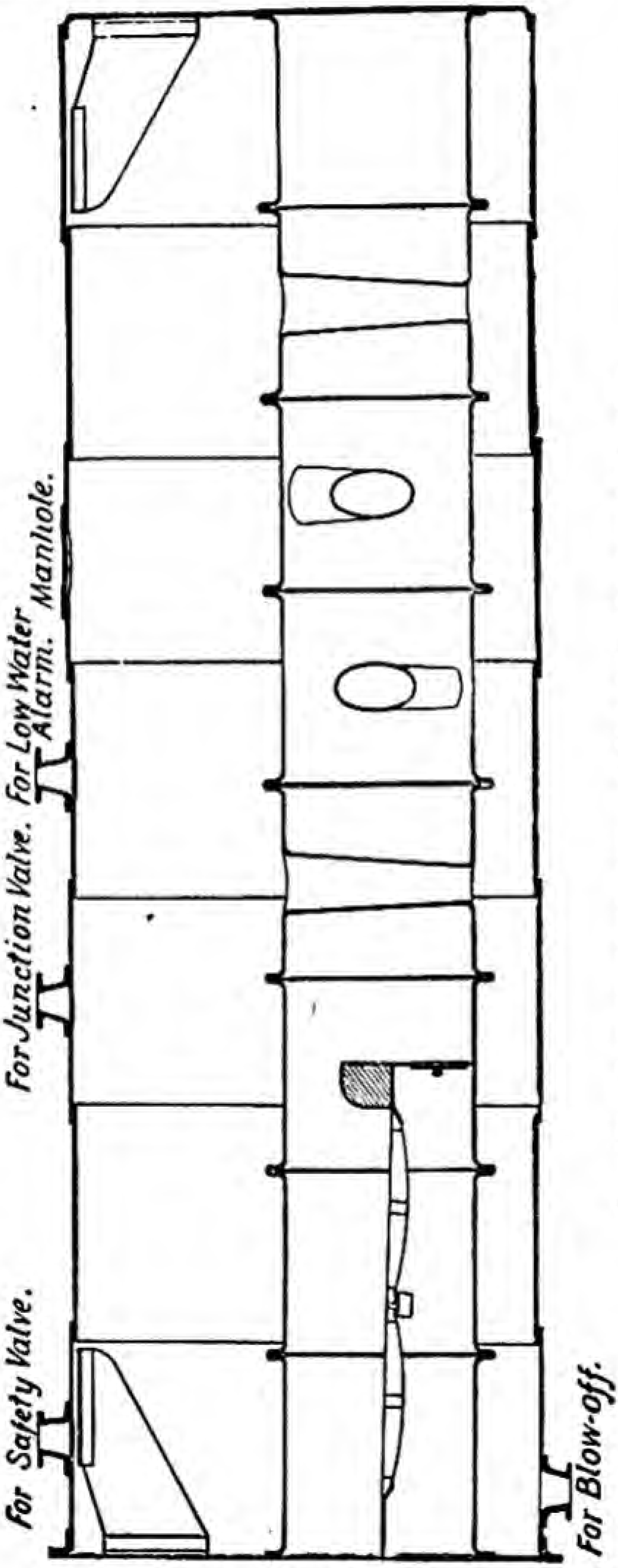


FIG. 135.—Longitudinal section of the shell of a Cornish boiler.

having direct access to the door and so rendering it too hot. The front grate bar support, called the **dead plate**, is made long

enough to prevent the fuel coming right forward to the furnace doors, thus protecting the joint of the furnace tube with the end plate from the direct action of the fire. The space below the fire grate is called the **ash pit**. The ash pit is closed at the back end below the fire bridge by a door, through which any ashes which may have found their way into the furnace tubes by falling over the bridge can be removed.

galloway tube.

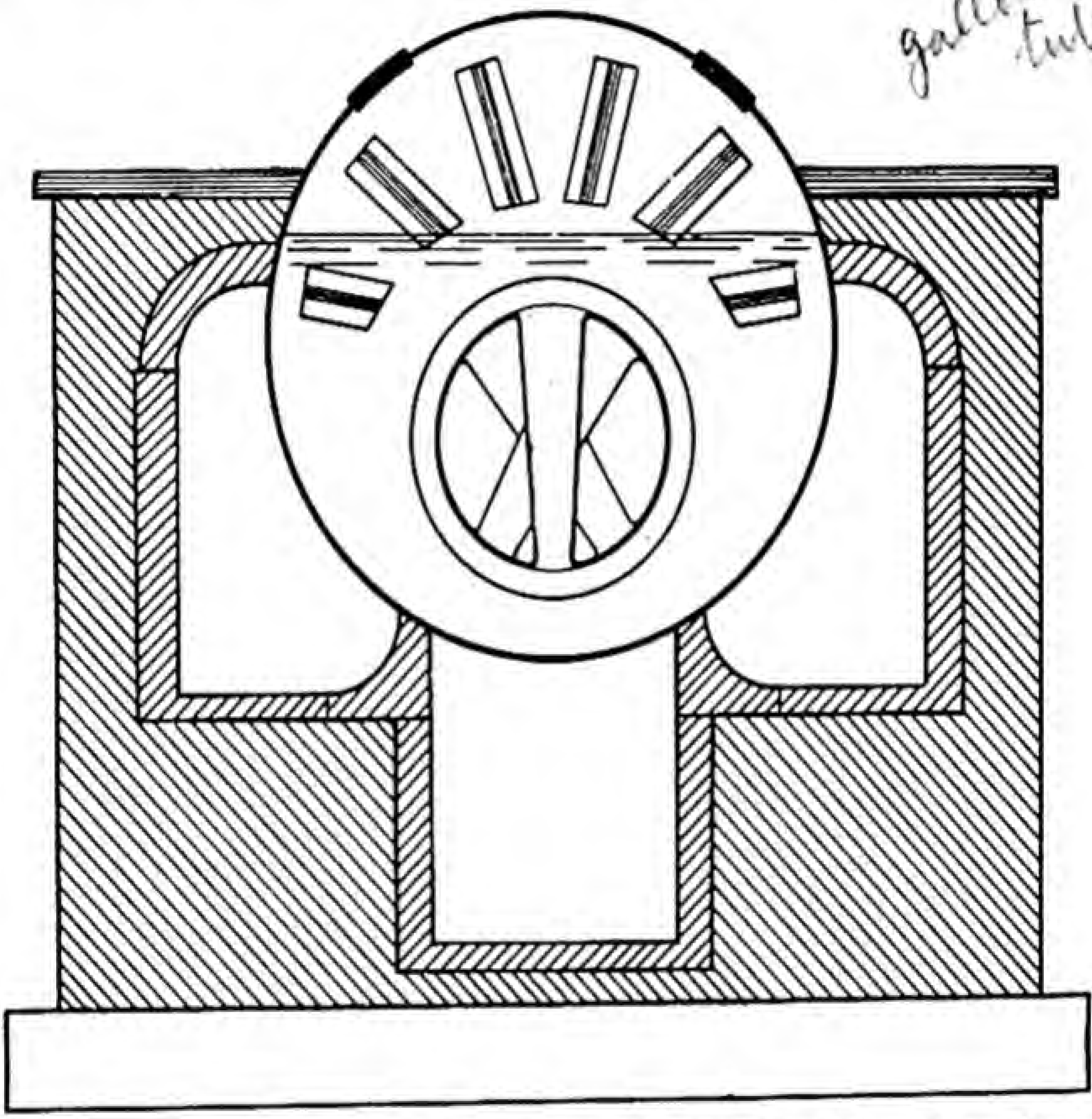


FIG. 135.—Cross section of a Cornish boiler and the brickwork seating and flues.

Access to the interior of the boiler is obtained through a man-hole placed on the top of the shell. A mud hole, situated on the front end plate near the bottom permits of the removal of deposits from the bottom of the boiler.

The Cornish boiler.—This boiler resembles the Lancashire boiler in all particulars, except that there is only one furnace tube. It is not usually made of so large a diameter—generally about 5′ 6″ to 6′ 0″. The diameter of the tube may be about 0·6 that of

shell. Fig. 135 shows a Cornish boiler in section, and in Fig. 136 is given a cross section through the boiler and brickwork. These drawings will be readily understood from what has been already said. The four cross tubes inserted in the furnace tube must be specially noted. These are called **Galloway tubes**, and their object is to improve the circulation of water in the boiler. This circulation is effected by the water in the tubes being heated by the furnace gases; its specific gravity being thereby lowered, an ascending current is set up in the tube, drawing away the cold water from the bottom of the boiler to be heated in turn. Lancashire boilers are also often fitted with Galloway tubes. The tubes are made conical in order that they may be got into place, and are welded at the top and bottom to the furnace tube; there are thus no rivets exposed to the fire action. Some of the tubes are vertical and others diagonal; the object of this is to break up the current of flue gases passing along the furnace tube, inducing the formation of eddies and promoting a scrubbing action of the gases on the walls of the tube. This arrangement greatly adds to the efficiency of the heating surfaces of the tubes, as every portion of the hot gases will, in turn, be brought into contact with plates having water to be heated on the other side.

Locomotive boiler.—The construction of a locomotive boiler will be understood by reference to Figs. 137 and 138, illustrating an express locomotive boiler constructed by the Great Eastern Railway Co. to the designs of Mr. James Holden.

The boiler consists of a cylindrical barrel *A* (Fig. 137), having an internal firebox *C* at one end, and a smoke box *O* at the other. The firebox is connected to the smoke box by 274 tubes $1\frac{3}{4}''$ external diameter. The furnace gases pass from the firebox through these tubes into the smoke box, and are discharged through the chimney *P*. The draught is obtained by discharging the exhaust steam from the cylinders through a blast pipe *Q* and a nozzle *R*, so situated in the smoke box as to induce a strong draught of air through the furnace and tubes. A movable cap is attached to the mouth of *Q*, and may be brought down so as to alter the area of the blast orifice and thus suit the conditions as to work being done by the engine. *M* is the steam dome, from the interior of which is taken the steam supply for the cylinders. A safety valve is mounted at *N*. The feed water is introduced through a valve at *Y*.

A large door *S* gives access to the smoke box and tubes for examination and cleaning.

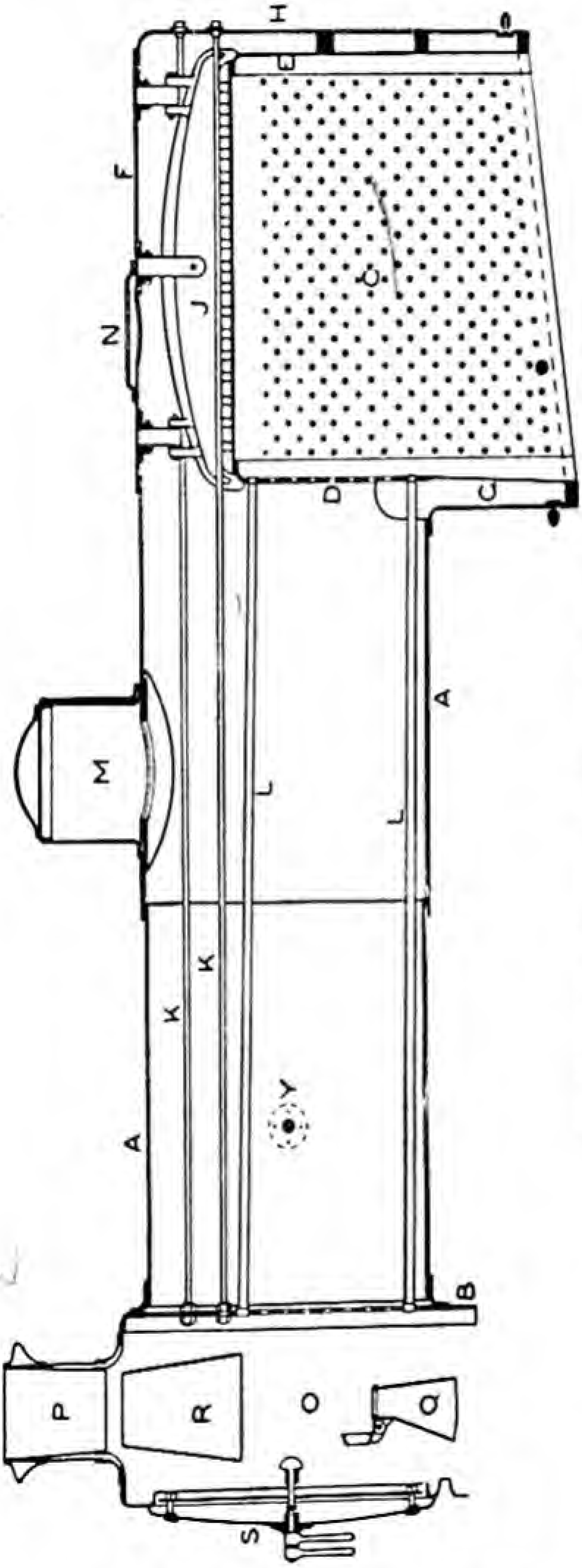


Fig. 137.—Longitudinal section of a locomotive boiler.

A. Barrel.
B. Front end plate.
C. Internal firebox.
D. Tube plate.
E. Firebox back plate.
F. Wrapper plate.
G. Throat plate.
H. Shell back plate.
J. Roof bar stays.
K. Longitudinal stays.
L. Tubes.
M. Steam dome.
N. For safety valve.
O. Smoke box.
P. Chimney.
Q. Blast pipe.
R. Nozzle.
S. Smoke box door.

Constructional details.—The barrel is 12′ 1″ long between the tube plates, and is built of two rings *AA* (Fig. 137), 4′ 8″ and 4 9″

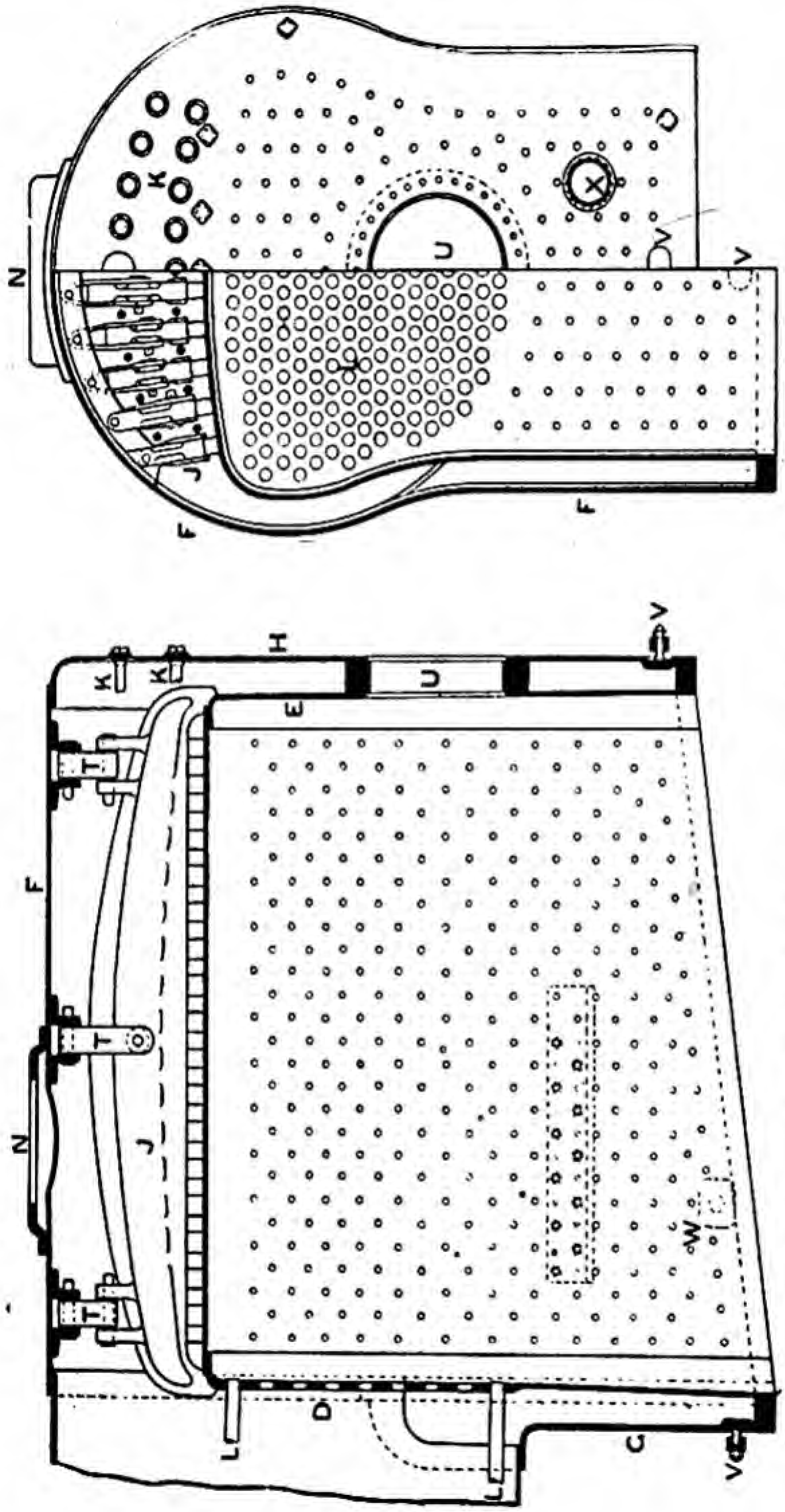


FIG. 138.—Longitudinal and cross sections of the firebox of a locomotive boiler.

T. Sling bars.
U. Fire door opening.
V. Hand holes.
W. For blow-off cock.
X. For liquid fuel nozzle.

diameter respectively, constructed of $\frac{1}{2}''$ plates. Each ring is made from a single plate; the longitudinal seams are butt joints with double cover straps and six rows of rivets; the circumferential seams are lap joints, single riveted. The front end plate *B* is connected to the barrel by an angle hoop, double riveted to the barrel and single riveted to the end plate. The plate is $\frac{5}{8}''$ thick, is flanged outwards to receive the smoke box, and is pierced with holes to receive the tubes. The shell casing round the firebox has a single wrapper plate *F* forming its sides and top (Fig. 138); *G* is a throat plate flanged to the barrel and to the wrapper plate. The back plate *H* is flanged to the wrapper, and is $\frac{9}{16}''$ thick. The steam dome *M* can be more clearly examined by reference to Fig. 152. The top is secured by means of studs so as to be removable for access.

Inside firebox.—This firebox is constructed of copper, a single wrapper plate forms the sides and top; the tube plate *D* and the back plate *E* are flanged into it (Fig. 138). The plates are $\frac{1}{2}''$ thick, excepting the tube plate which is 1″ thick at the tubes. Connection to the outer shell is made round the bottom edge, a distance piece of rectangular section being inserted, through which the rivets pass. The fire door opening is at *U*; *V*, *V* are hand holes; the blow-off cock is secured to the boiler at *W*. *X* is one of two orifices to which the nozzles for burning liquid fuel are attached; the nozzles will be more fully dealt with in Chap. XVI.

Stays.—The sides of the firebox are stayed to the outer wrapper plate, the front plate to the throat plate, and the firebox back plate to the shell back plate by means of a large number of bronze stays, $\frac{15}{16}''$ diameter, screwed into both plates and riveted over. The roof of the firebox is supported by ten roof bars *J* (Fig. 138), of I section, the ends of which rest on the edges of the firebox. Sling bars *T* connect the roof bars to the outer shell; the firebox roof is connected to the roof bars by means of bolts screwed into bosses on the under side of the roof bars. The front tube plate of the boiler is stayed to the back plate by means of longitudinal stays *K*, passing from end to end of the boiler and secured by outside nuts screwed up against washers. These stays are $1\frac{1}{8}''$ in diameter.

The working pressure of this boiler is 180 lbs. per square inch. The total heating surface is 1630 square feet; the grate area is 21·3 square feet.

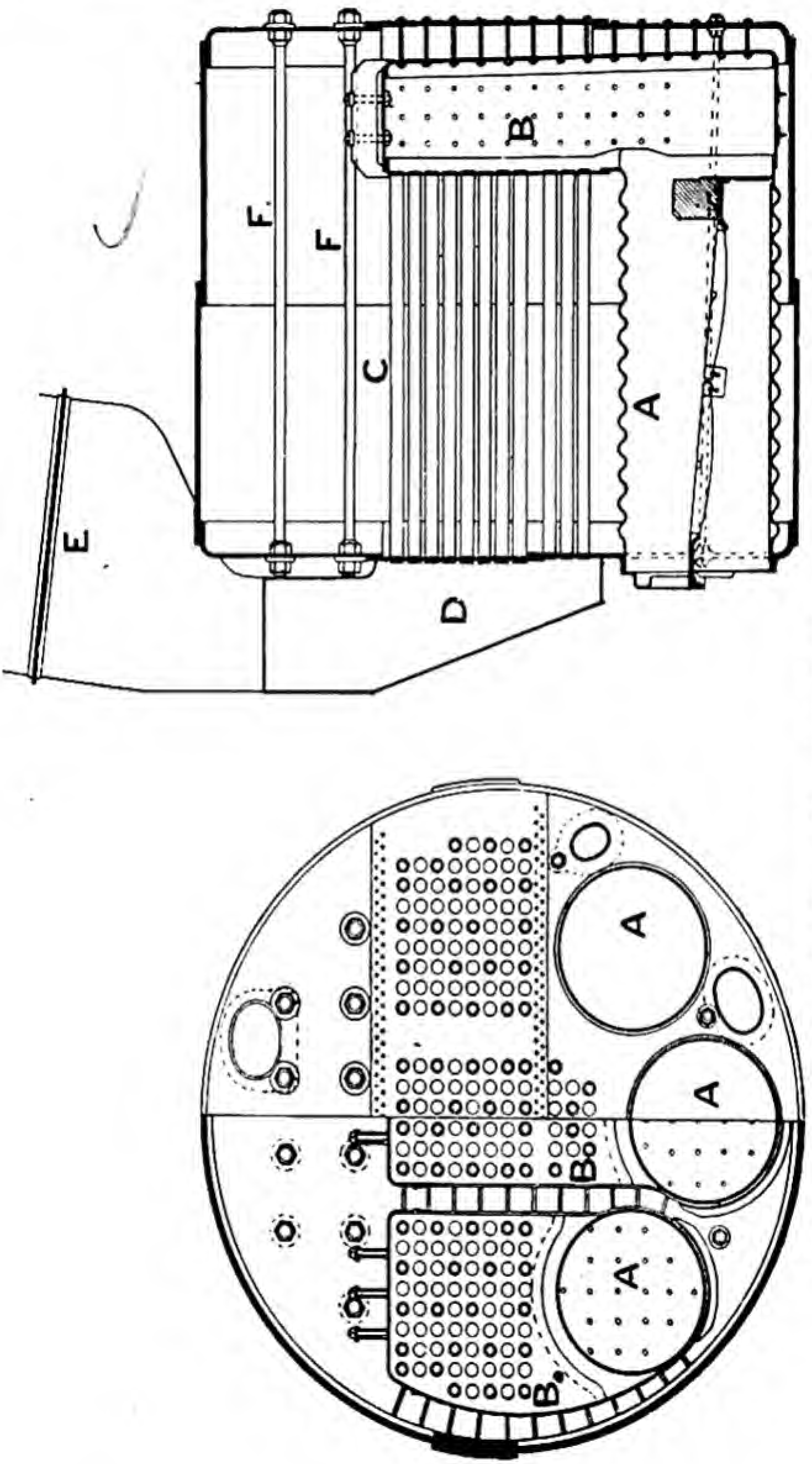


FIG. 139—Sectional front elevation and longitudinal section of a return-tube marine boiler.

A. Furnace tubes.
B. Combustion chambers.
C. Tubes.
D. Uptake.
E. Chimney.
F. Longitudinal stays.

Marine return-tube boiler.—This boiler consists of a cylindrical shell fitted with from two to eight furnaces, and having a large number of tubes passing from internal combustion chambers to the front plates of the shell. In **single-ended boilers** of this type the furnaces are at one end of the shell only; in **double ended boilers** there are furnaces at both ends. The gases from the furnaces pass into the combustion chambers, then return through the tubes to the front end in single-ended boilers, or to both ends in double-ended boilers. Uptake passages are provided leading from the boiler ends to the funnel. As the furnace gases pass through the tubes, this is a fire-tube boiler.

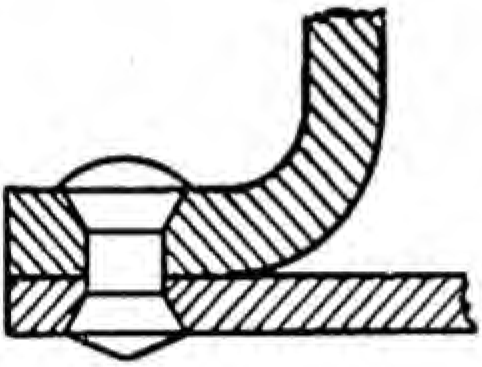


FIG. 140.—Connection of furnace tube to front plate in a marine boiler.

Construction of return-tube boiler.—The drawings in Fig. 139 show a single-ended boiler constructed by the Thames Engineering Works, Ltd., Greenwich. There are three furnaces contained in corrugated tubes, *A*, the smaller internal diameter at the bottom of the corrugations being 3′ 0″ and the larger internal diameter being 3′ 3″. The corrugations are 6″ pitch and the tubes are $\frac{17}{32}$″ thick. The furnace tubes are riveted to the front end plate, which is flanged outwards to receive them (Fig. 140), and at the inner ends to the combustion chambers (Fig. 141).

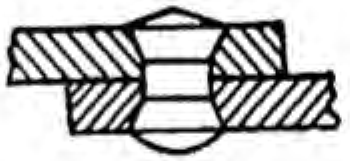


FIG. 141.—Combustion chamber joint.

There are three combustion chambers, *B*, a centre one and two **wing** chambers. These are somewhat rectangular in shape, and are strongly constructed in order to withstand the external collapsing pressure. The front plates, which receive the tubes, are $\frac{13}{16}$″ thick, the top plates $\frac{5}{8}$″, and the side and back plates $\frac{1}{2}$″. The back plate slopes slightly forward in order to allow the steam bubbles forming on the plate to disengage themselves instead of creeping up the plate. The back of each combustion chamber is stayed to the back end of the boiler by a large number of screwed stays ranging from $1\frac{1}{4}$″ to $1\frac{5}{8}$″ diameter placed about $6\frac{3}{4}$″ pitch. The sides of the wing chambers nearest to the outer shell are similarly stayed thereto. The inner sides of the wing furnaces are stayed

in the same manner to the sides of the centre chamber. The combustion chamber tops are supported by girder stays (Fig. 142) consisting of two plates $6\frac{1}{4}''$ deep and $\frac{3}{4}''$ thick riveted together and

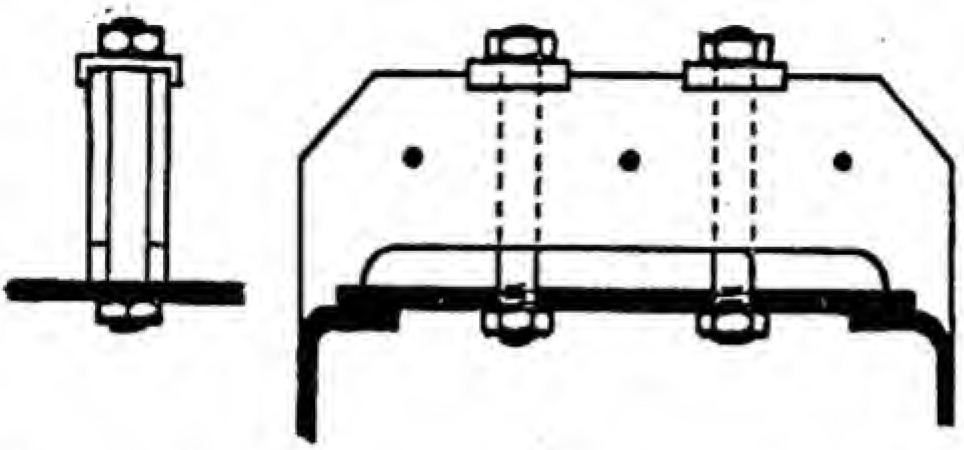


FIG. 142.—Girder stays supporting the combustion chamber roofs.

resting on the upper edges of the combustion chambers. Steel stays, $1\frac{5}{8}''$ diameter, with top and bottom nuts, secure the combustion chamber tops to the girder stays. The tube plates of the combustion chambers are stayed to the front end of the boiler by **stay tubes**. Most of the tubes, which are $3\frac{1}{4}''$ external diameter, are simply pushed into place and then expanded at the tube plates to make a steam-tight joint. Others, shown by a double circle in the front elevation (Fig. 139), and in detail in Fig. 143, are screwed at both ends, and the holes in the front end of the shell and in the tube plates of the combustion chambers are screwed to receive them; these tubes effectually stay the flat tube plates. The combustion chamber bottoms are strengthened by means of T's, riveted to the outside of the chambers.

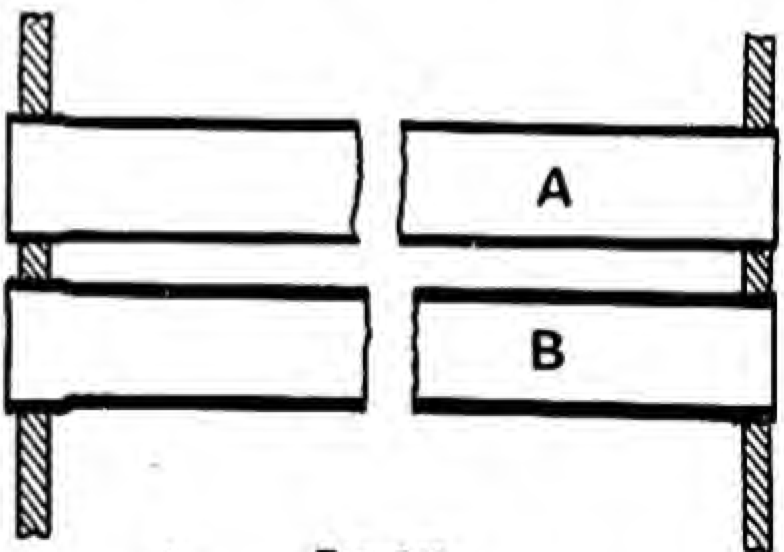


FIG. 143.

A. Plain tube. B. Screwed stay tube.

The shell of the boiler is 12′ 3″ mean diameter and is constructed of plates $1\frac{1}{8}''$ thick. There are two rings, each ring being constructed of two plates, with butt joints so as to give a perfect cylinder. The longitudinal joints have double cover plates, and

are secured by six rows of rivets. The circumferential joints are lap, with double row of zig-zag rivets.

The ends of the boiler are each constructed of three plates, with horizontal lap joints double riveted. The top plates, front and back, are each 1″ thick, the other front plates are $\frac{7}{8}$″, and the back plates $\frac{13}{16}$″. The ends are stayed, where unsupported in the manner previously described, by means of longitudinal stays *F* (Fig. 139), $2\frac{5}{8}$″ diameter in the body, swelled at the ends, where they are screwed, to 3″ diameter (Fig. 144). A large washer, 7″ diameter outside, and nuts both inside and outside, secure the stay to the plates. The washer serves to distribute the pull over the neighbouring part of the plate. These stays are pitched 1′ 5″ in the upper part of the shell, and there is one between the centre furnace tube and each wing tube, and another just above each wing tube, the last mentioned stays running from the front end plate to the combustion chamber tube plate.

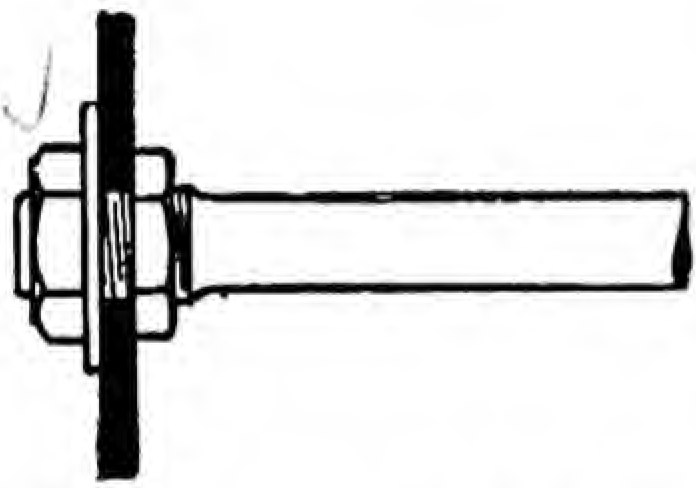


FIG. 144.—End of a longitudinal stay.

D is the uptake, leading to the funnel *E* (Fig. 139).

The furnaces are shown in more detail in Fig. 145. The grate is 6′ 0″ long, made up of two lengths of fire-bars *C*, resting on the dead plate *D*, on centre bearers *E* and on the bridge support *F*. A thin ash plate rests on the bottom of the ash pit to facilitate the removal of ashes which, owing to the corrugations, would otherwise be difficult. The ash pit is fitted with a swing door *J*, providing a supply of air, which may be adjusted by varying the door opening. The door is held open by the notches cut in a swing bar. A plan of some of the fire-bars is shown in Fig. 145. The air spaces between the bars are $\frac{5}{8}$″ wide and $14\frac{1}{2}$″ long. The bars are 1″ wide at the narrow part.

The boiler is constructed for a working pressure of 160 lbs. per square inch, and to be tested to 320 lbs. per square inch. The total heating surface is 1500 square feet; the total grate area is $56\frac{1}{4}$ square feet; the steam space is 245·5 cubic feet. There are 47 plain tubes and 22 stay tubes in each wing, and 38 plain tubes

and 24 stay tubes in the centre, giving a total of 200 tubes, each $3\frac{1}{4}''$ external diameter and 7′ 1″ long between the tube plates. The stay tubes are $\frac{1}{4}''$ thick, the others are No. 8 B.W.G. The front circumferential seam of the shell is hand riveted, the remainder of the joints being machine riveted as far as possible.

Access to the boiler is obtained by three manholes placed on the front end, one near the top and one between the centre and each wing furnace. A sight hole is placed just outside each wing furnace. These are all fitted with M'Neil's patent doors.

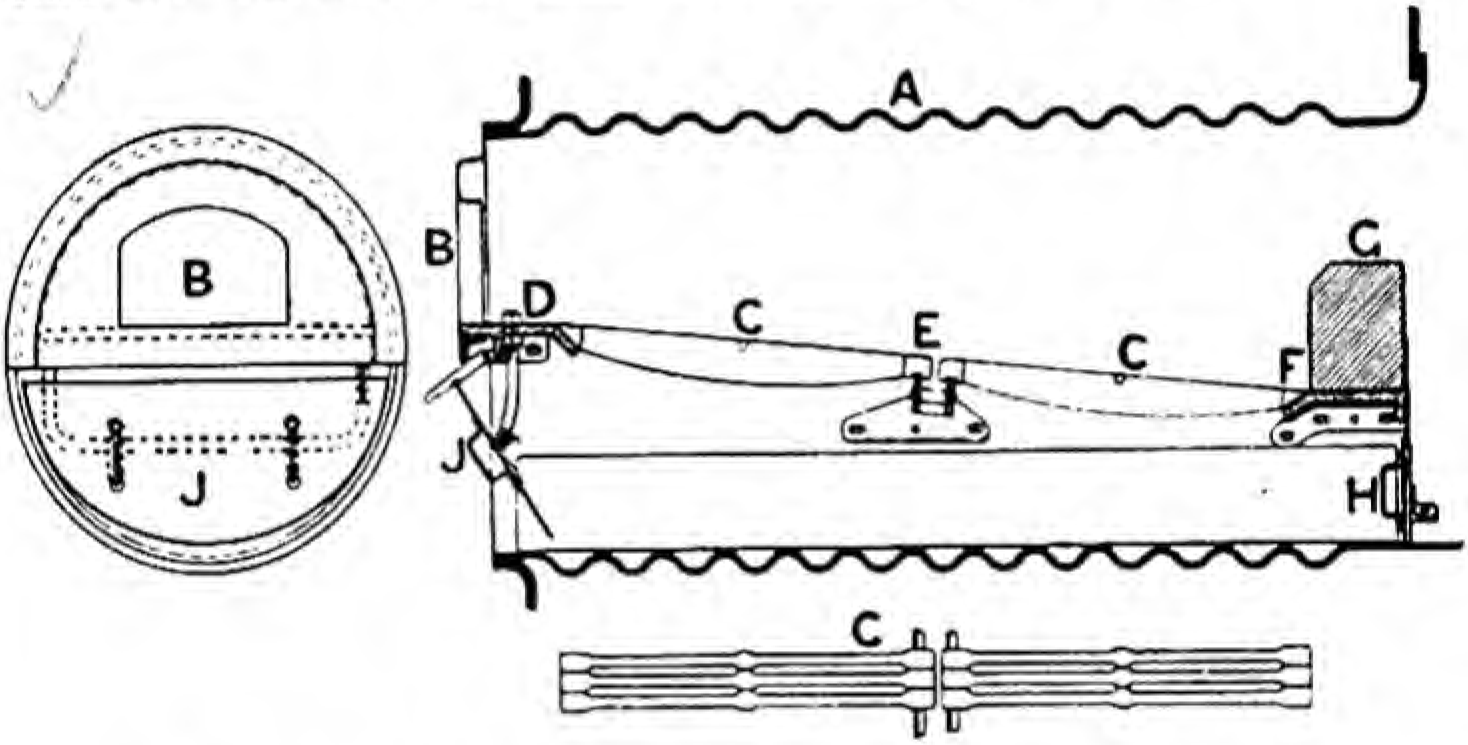


FIG. 145.—Front elevation and section of a marine furnace, with plan of some of the fire bars.

Arrangement of marine boilers.—Two boilers similar to that described are required for the triple expansion engine described in Chapter XV. The I.H.P. to be supplied is about 1350. Fig. 146 shows the boilers arranged in position in the ship; the principal mountings are indicated in this illustration. *D*, *D* are the main steam pipes and valves supplying steam to the engines; the valves are 5″ in diameter. *E*, *E* are $2\frac{1}{2}''$ stop valves supplying steam to the auxiliary machinery. *C*, *C* are double spring safety valves, $2\frac{3}{4}''$ in diameter, and fitted with escape pipes. *A*, *A* are brackets on which are mounted the water-gauge cocks. The brackets are connected to the steam and water spaces in the boiler by $1\frac{1}{2}''$ copper pipes, a valve being placed where each pipe enters the boiler. This construction is rendered necessary in order to clear the uptakes. *B*, *B* are test cocks for roughly checking the water level in the boiler. *H*, *H* are feed check valves through which

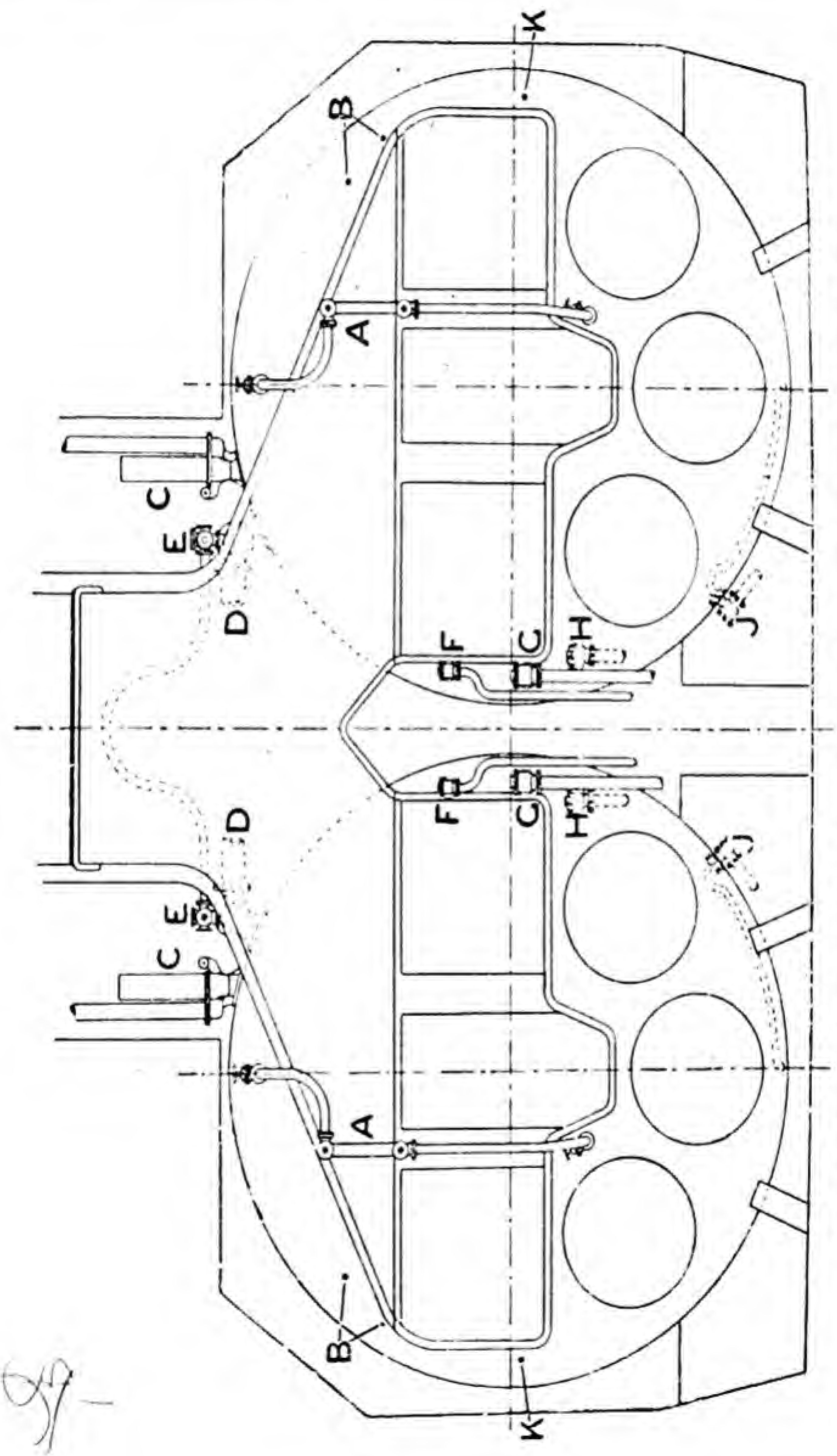


FIG. 146.—View, looking forward, of two marine boilers mounted in position on board ship.

the water from the feed pumps driven by the engine enters the boilers. These are situated on the back end. *G*, *G* are similar valves on the front end through which water from the auxiliary feed pumps is forced into the boiler. The feed valves are all $2\frac{1}{4}''$ in diameter. *F*, *F* are **scum cocks** connected by an internal pipe to a **scum dish** just below the working water level in the boiler. These serve to drain off scum from the surface of the water in the boiler. *J*, *J* are 2″ blow-off valves, fitted with internal pipes

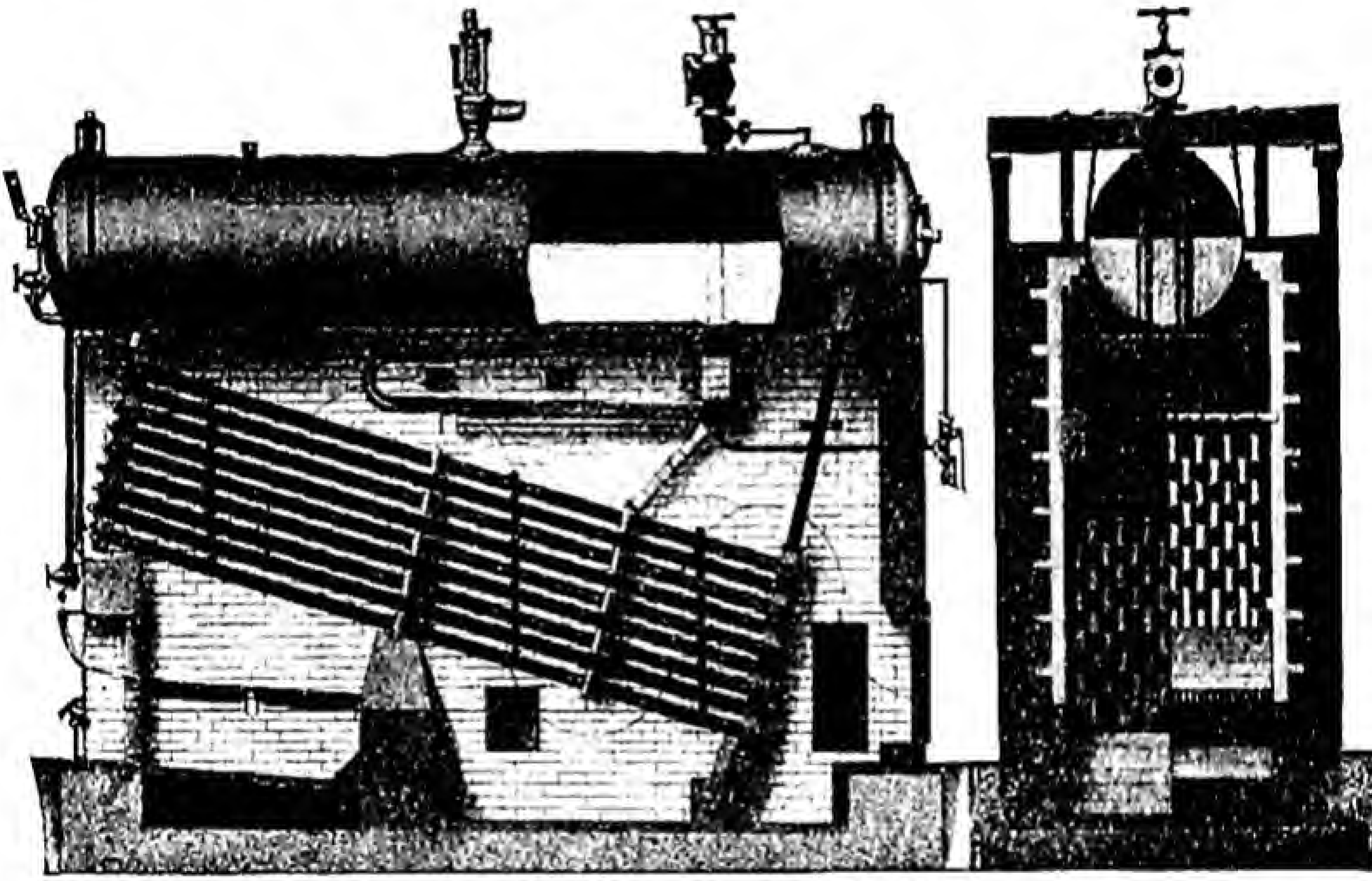


FIG. 147.—Longitudinal and cross sections of a Babcock & Wilcox water-tube boiler and its seating.

reaching to the lowest level in the boiler. Similar valves are fitted to each boiler and connected to the auxiliary pumps, thus enabling the water in the main boilers to be pumped out when cold. *K*, *K* are **salinometer cocks** used in testing the degree of saltness of the water in the boiler. To the top of each boiler a pressure gauge cock and an air cock are fixed. The boilers rest on strong brackets carried by the ship's framing; the space surrounding the boiler-room is taken up with coal storage bunkers.

Water-tube boilers.—Space will not permit of the description of more than one boiler of this type, and the well known **Babcock & Wilcox boiler** has been selected.

This boiler consists essentially of three parts, shown in Figs. 147, 148, and 150.

(i) A number of **inclined water tubes** over the furnace, in which the water, being divided into small volumes, is raised to a high

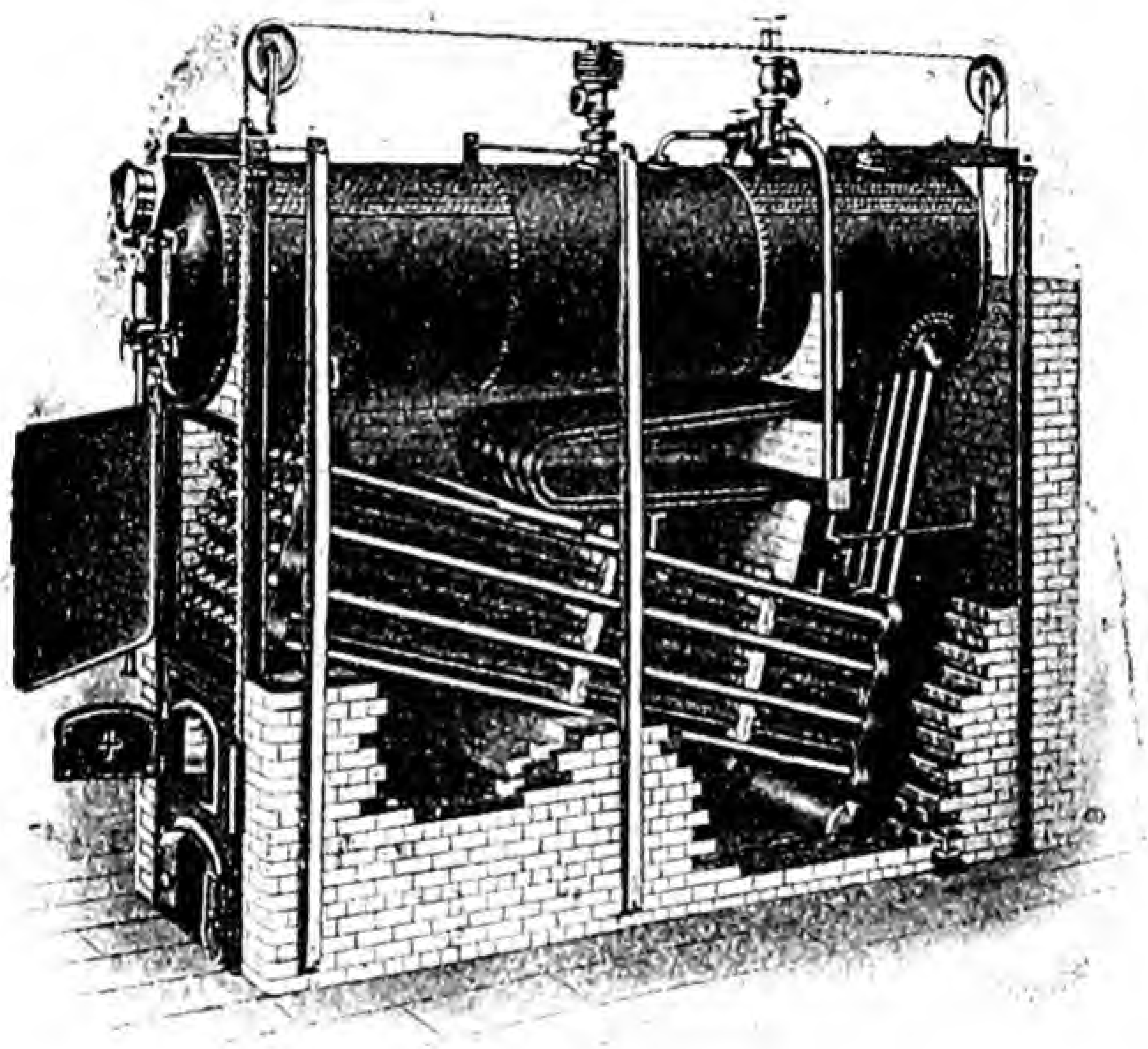


FIG. 148.—Perspective view of a Babcock & Wilcox boiler with brickwork partly removed to show the tubes, mud collector, and superheater.

temperature quickly and rises through vertical connecting boxes or **headers,** to which the front ends of the tubes are connected, into

(ii) A **horizontal steam and water drum.** In this drum the steam separates from the water (Fig. 150). The water remaining travels to the back end of the drum and descends through vertical tubes into the inclined water tubes where it is subjected to the action of the fire, and again passes into the steam and water drum. Thus, a continuous and rapid circulation of the water is kept up, and approximately uniform temperature preserved throughout the boiler.

(iii) **A mud collector** attached to the lowest part of the inclined water tubes (Fig. 148). Any sediment, by reason of its greater specific gravity, will be precipitated into this collector during the passage of the water through the rear headers.

The holes in the headers for the reception of the water tubes are **staggered** (Fig. 149) so that each row of tubes comes over the spaces in the next lower row. Hand holes are provided in the headers opposite each tube, thus enabling each tube to be cleaned out and examined. The steam and water drum has its ends domed, thus obviating the necessity for stays.

FIG. 149.—Header, showing the staggered holes for receiving the tubes.

Seating.—The boiler is erected entirely independent of the brickwork, being suspended from wrought-iron girders resting on iron columns as will be observed in the end elevation in Fig. 147. This construction prevents unequal expansion troubles, and facilitates repairs to the brickwork. The grate is under the front and higher ends of the water tubes. The furnace gases are compelled by **baffle plates** (Fig. 147, longitudinal section) to pass upwards between the tubes into a combustion chamber under the steam and water drum, thence downwards between the tubes, then once more upwards between the tubes and off to the chimney. The hottest gases thus come in contact with the highest parts of the water tubes wherein the hottest water will be found. A damper at the rear flue opening to the chimney is operated by means of a chain led to the boiler front, and serves to regulate the draught.

The feed water is introduced through a feed valve on the front end of the steam and water drum, and is directed backwards along the drum by a short internal pipe, this being the natural direction of flow. The valve through which the steam leaves the boiler is situated near the rear end of the steam and water drum, the steam liberated near the front headers has thus to travel a considerable distance along the drum before making its exit, thus promoting the production of dry steam.

Superheater.—The boiler illustrated in Figs. 147 and 148 is fitted with a **superheater**. The superheater consists of a number of

horizontal U-tubes secured at each end to horizontal connecting boxes, and placed in the combustion chamber just under the steam and water drum. Steam is led into these tubes by means of a vertical T-tube, the upper branches of the T being situated in the

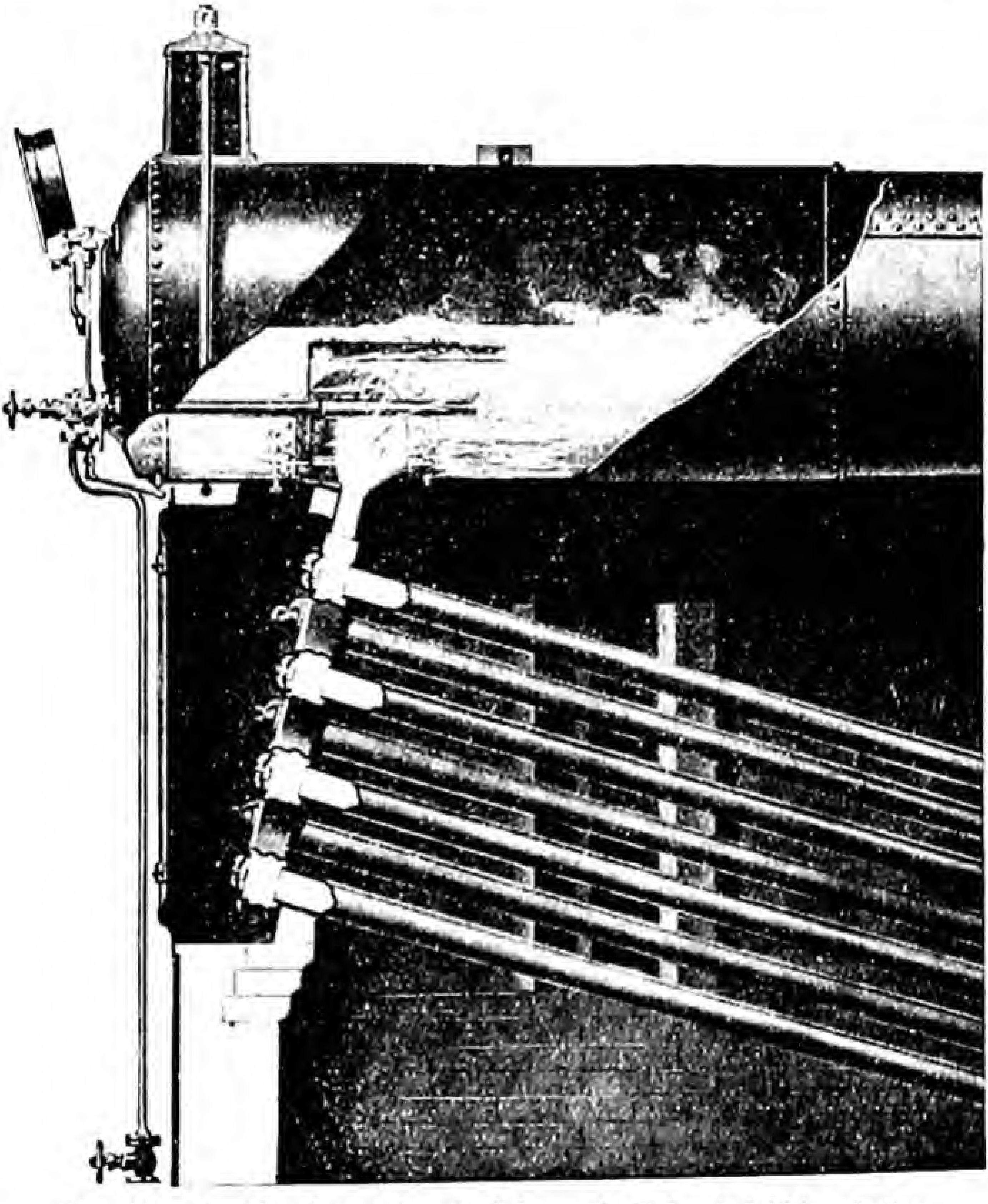


FIG. 150.—Front headers and part of drum of a Babcock & Wilcox boiler, showing the water circulation.

steam space of the drum. The steam, when the superheater is in use, leaves the drum through the T-tube, passes into the upper connecting box of the superheater, thence through the U-tubes, attaining a high temperature as it does so, and is finally drawn from the lower connecting box into the main steam pipe supplying the engine.

Heating surface and grate area.—The power of a given boiler may be estimated by stating the quantity of water which can be evaporated per hour. This quantity will evidently depend on the quantity of coal which can be burned per hour in the furnaces, and also on the extent of the heating surface and the suitability of its arrangement. Given the area of the fire-grate and the strength of the draught, a definite quantity of coal can be burned per hour. Engineers have been led by experience to provide a certain ratio of heating surface to grate area, the value of the ratio depending on the type of boiler. For example, the value of the ratio in the Great Eastern Railway locomotive boiler (p. 164) is $1630 \div 21{\cdot}3 = 76{\cdot}5$, and in the marine return tube boiler (p. 168), the ratio is $1500 \div 56\frac{1}{4} = 27$ nearly. The explanation of the difference in these ratios lies in the fact that the rate of combustion, *i.e.* the weight of coal burned per square foot of grate area per hour, will be much higher in the locomotive boiler than in the marine boiler. A larger proportion of heat being thus available per square foot of grate in the locomotive boiler, a larger amount of heating surface must be provided in order to ensure that this heat may be efficiently passed into the water in the boiler.

EXERCISES ON CHAPTER X.

1. Give sketches and describe the arrangements of the brickwork flues of a Lancashire or Cornish boiler. Indicate the flow of the furnace gases by arrows.

2. Sketch and describe the shell of a Lancashire boiler. Give separate sketches of the joints of the shell plates and of the attachments of the ends.

3. Describe and give sketches of the construction of the furnace tubes of a Lancashire or Cornish boiler. Explain how the successive sections are connected and how the tube is fastened to the end plates. Why are Galloway tubes frequently fitted?

4. Give sketches and description of the furnace arrangements of any boiler you know.

5. Sketch and describe the shell of a locomotive boiler.

6. Give sketches of four different kinds of stays, explaining the part of the boiler for which each is suitable.

7. Sketch and describe the shell of a return-tube marine boiler.

8. In what circumstances are water-tube boilers used in preference to fire-tube boilers? Give reasons for your answer.

9. Give sketches showing the arrangement in any water-tube boiler. Do not show details of joints or fittings, but indicate the course of the furnace gases and the water circulation by arrows.

10. Give sketches and description of any type of self-contained vertical boiler. Do not show details.

11. Describe, with sketches, the fire-box of a locomotive, showing how it is stayed. Give larger sketches of a few details. 1907.

12. Describe, with sketches, any important part of any water-tube boiler. 1907.

CHAPTER XI.

BOILER MOUNTINGS.

Junction valve.—This valve is bolted to a mounting block on the top of the boiler; and through it passes the principal supply of steam from the boiler. The valve shown in section in Fig. 151 is designed to obviate, as far as possible, trouble due to unequal expansion in its parts. The body of the valve, *A*, is strongly made of cast-iron for pressures up to 160 lbs. per square inch, and of Siemens-Martin cast-steel of mild quality for higher pressures. The valve *B* is of gun-metal and is in one piece with its spindle *C*, which is bored out to receive another steel spindle *D*. A gun-metal sleeve *E*, screwed to fit a tapped hole in the cross bar *F*, is drilled to permit *D* to pass through it, and is enlarged at its lower end to accommodate the upper end of the valve spindle and also a collar on the spindle *D*. The spindle *D* and the valve spindle *C* are connected by

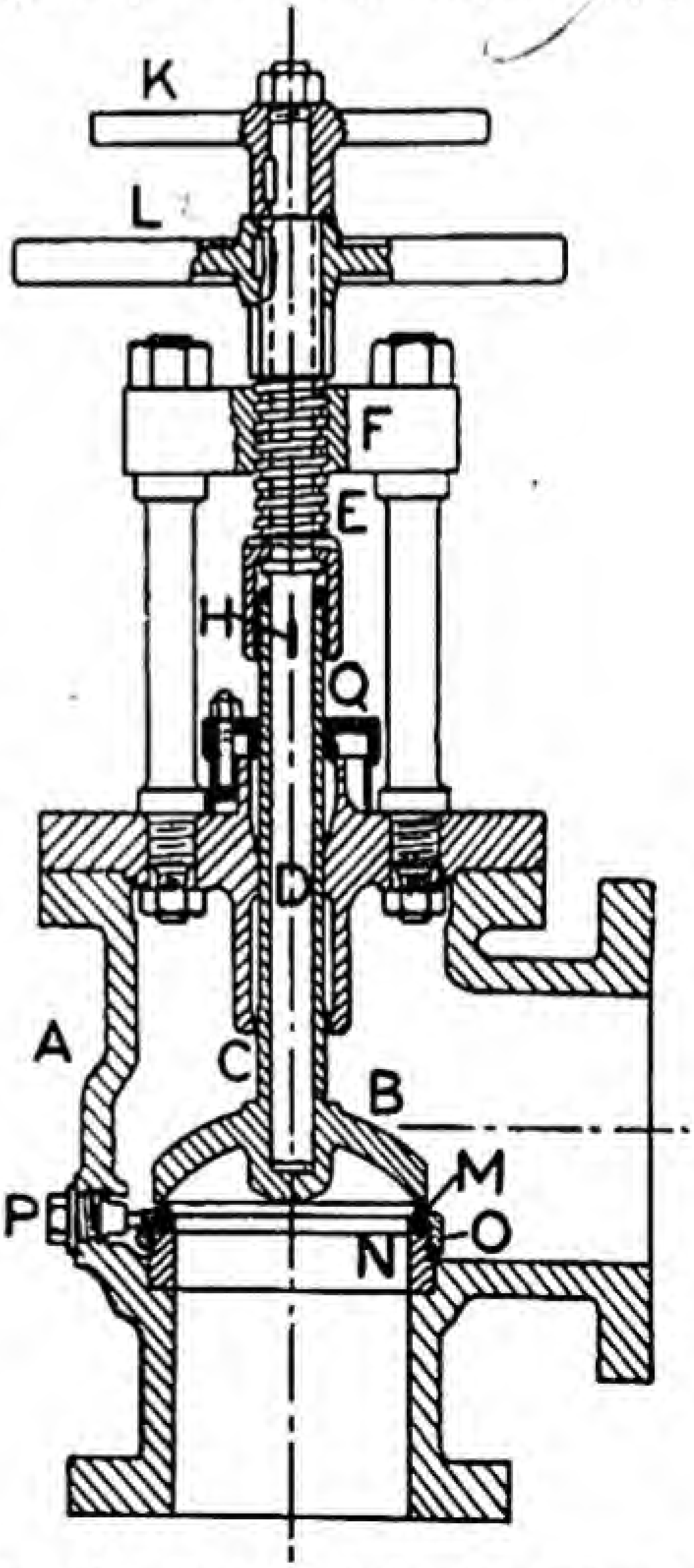


Fig. 151.—Hopkinson's junction valve for main steam supply.

means of a cotter *H*; so that, if *D* be rotated by the cross bar *K* secured to its upper end, the valve will also be rotated. Rotation of the hand wheel *L*, which is secured to the sleeve *E*, will raise or lower the valve and so open or close it, but will not produce rotation of the valve. This construction, it should be observed, leaves the valve spindle free to expand lengthways, and as its expansion will be greater than that of the valve body, the valve will not thereby be jammed to its seat.

The valve seat, *M*, is held against a ring *N* by a loosely fitting gun-metal ring *O*, screwed to fit *M*. A set screw *P* serves to keep the whole in place. The arrangement permits of free expansion to the seat, *M*, which therefore will not warp when subjected to changes of temperature. A stuffing-box *Q* serves to keep the valve spindle tight against steam leakage.

Anti-priming pipe.—The position of the junction valve on the boiler will be seen by reference to Fig. 121. The supply of steam is taken from the boiler through an **anti-priming pipe** placed inside the boiler. This pipe is 4 or 5 feet long with closed ends, and has a number of perforations on its top side (Fig. 123, *L*). A branch connects it with the junction valve mounting block. As its name implies, its function is to reduce the quantity of water carried over with the steam through priming. Priming is generally caused by a too rapid ebullition of the water and by a too restricted steam space in the boiler. Drops thrown above the surface of the boiling water and caught in the currents of steam travelling towards the exit are "baffled off" by the anti-priming pipe.

Locomotive regulator.—In locomotives, the valve through which the steam passes from the boiler to the cylinders, called a **regulator**, is generally placed inside the boiler. A dome *A* (Fig. 152) is riveted to the top of the boiler, and the regulator is placed within it so as to draw as nearly dry steam as possible. Usually the regulator consists of two slide valves *B* and *C*, one operating on the back of the other, and worked from a handle *D* at the back plate of the boiler, to which the valves are connected by rods and levers. The outer valve *C* is smaller than the inner, and opens first, admitting steam through a small hole in the inner valve. Steam being thus admitted to the steam pipe *F*, leading to the cylinders, the pressures on the opposite sides of the large inner valve will be

equilibrated, and this valve may now be operated easily. To provide for the required movement, the pin *E*, on the rod operating the valves, is a working fit in a hole formed in the outer valve, but engages in a slotted hole in the inner valve.

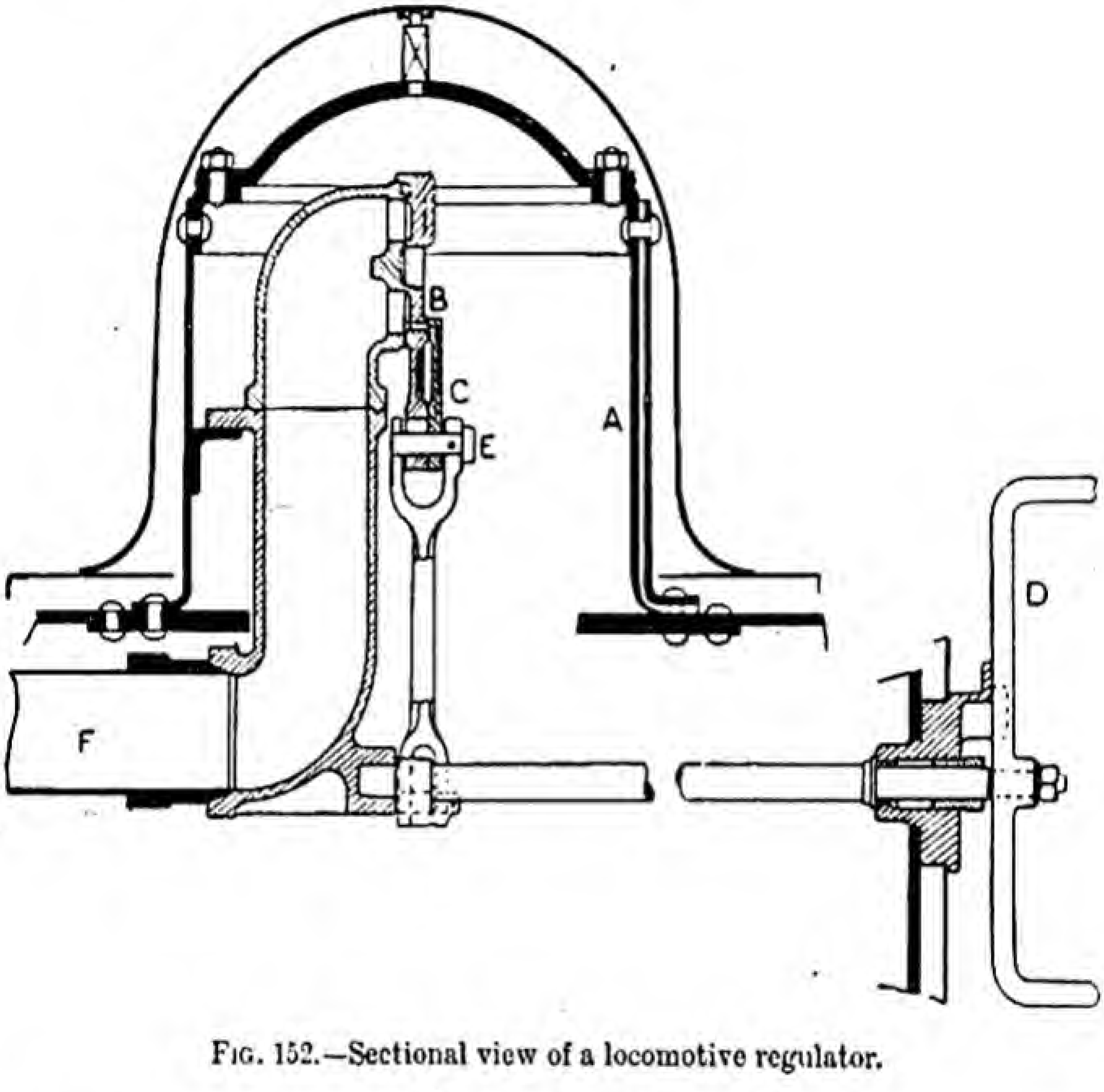


FIG. 152.—Sectional view of a locomotive regulator.

Marine automatic stop valve.—This stop valve is intended for each of the boilers shown in Fig. 146. Steam passes from the boiler through the valve into a common steam main. The valve is so constructed as to close automatically should any accident occur to the boiler on which it is mounted, thereby localising the extent of the damage and permitting the other boiler to continue to supply steam. Referring to Fig. 153, showing the valve in section and plan, *A* is the valve made in a continuous piece with a spindle *B*. A cross bar *C* attached to the top of *B* enables the valve to be rotated on its seat. The valve is held down by a sleeve *D*, bearing against a shoulder on the spindle *B*, and screwed externally to fit a

tapped hole in a fixed bridge, a portion of which is shown at *E*. The sleeve *D* has attached to it a handwheel *F*, by means of which it may be rotated, and so raised. The steam coming from the boiler at *G* is now able by its pressure to raise the valve *A* and so to pass through *H* into the steam main. Should any accident occur to the boiler, thus reducing the pressure on the side *G* of the valve, the higher pressure in the steam main will at once cause the valve to close and thereby to isolate the boiler. *J* is a branch fitted with another stop valve, not shown in Fig. 153, by means of which steam is supplied to the auxiliary machinery.

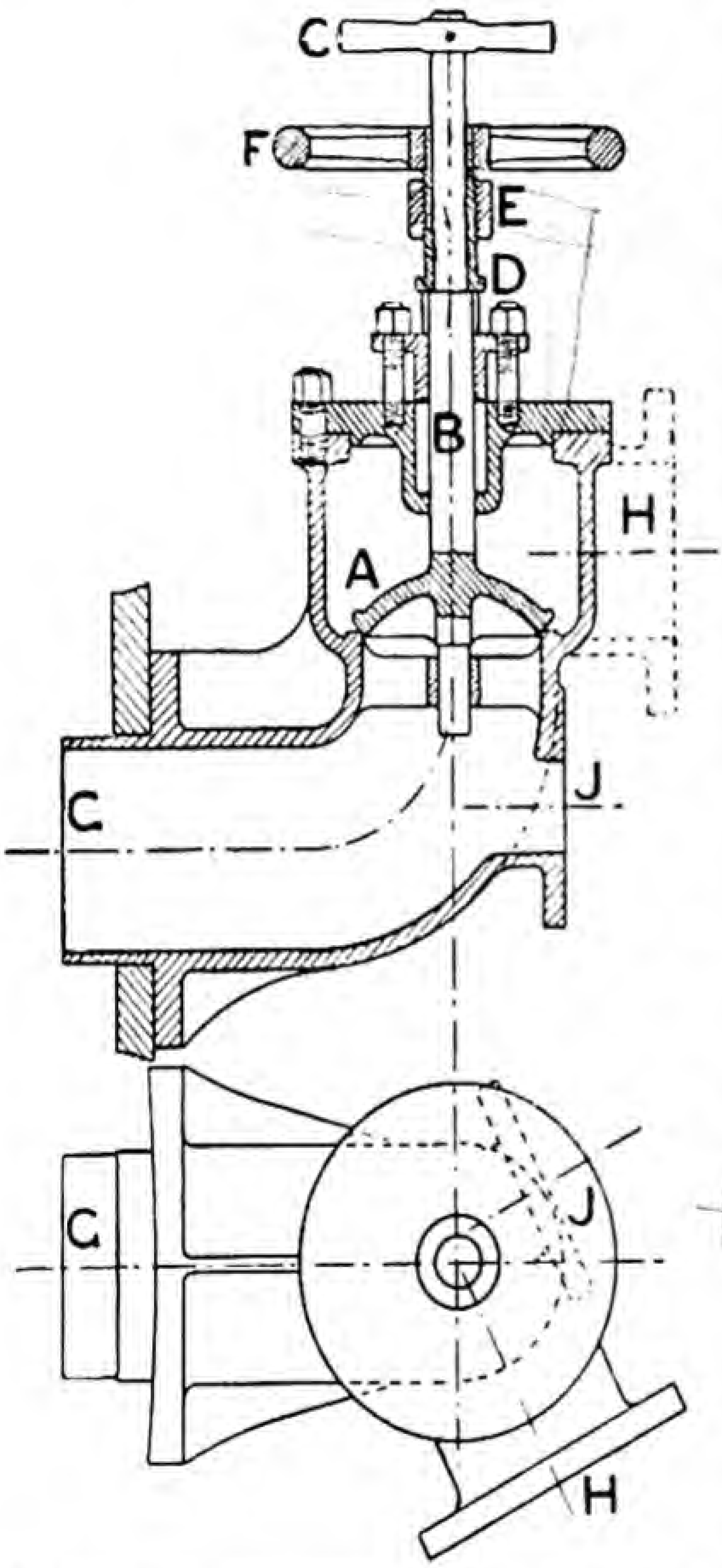


FIG. 153.—Sectional elevation and plan of a marine automatic stop valve.

Water gauge.—The object of this fitting is to indicate the level of the water in the boiler and so to enable the attendant to regulate the supply of feed water. The gauge is connected to the boiler by means of flanges at *XX* (Fig. 154), the upper opening into the steam space and the lower into the water space in the boiler. *G* is a strong glass tube, passing through stuffing-boxes into the upper and lower parts of the gauge. *D* and *E* are steam and water cocks respectively, and,

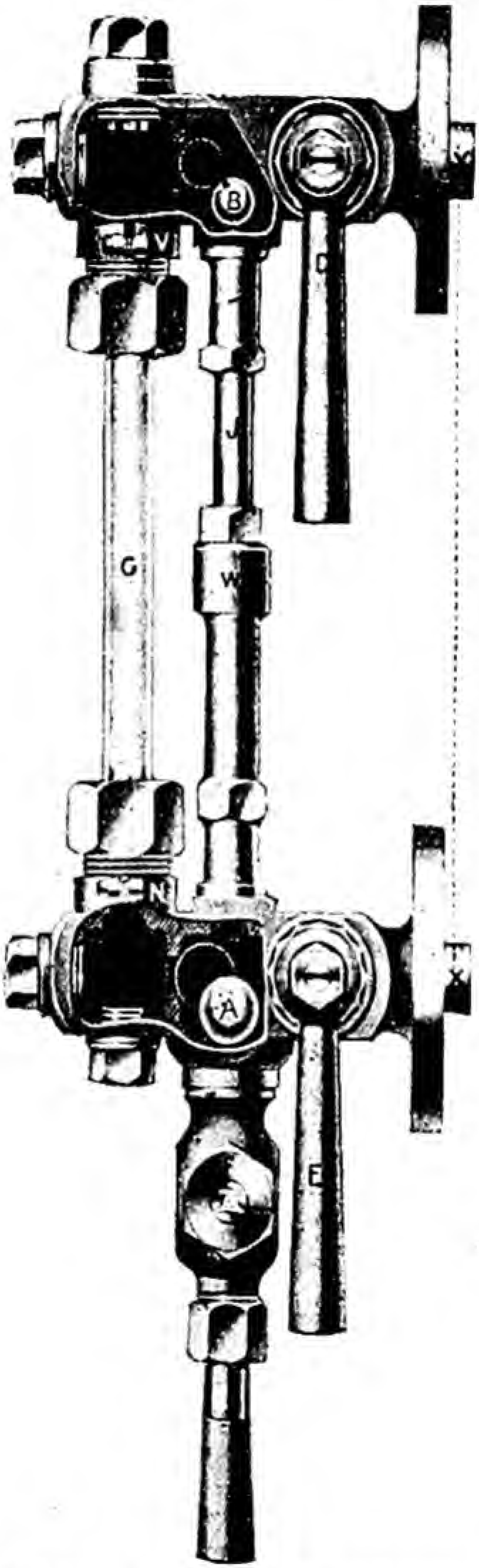


FIG. 154.—Hopkinson's water gauge with automatic cut off valves.

when open, allow water from the boiler to flow into G until the same level is attained in the tube as that existing in the boiler. F is a drain cock, by means of which the whole of the fitting can be blown through. On closing F, the water should rise freely and rapidly to its original level. Should there be any signs of blocking, the cocks E and D should be closed, and the gauge examined and cleared. There are four small screwed plugs shown; these being removed, a wire may be pushed horizontally through the top and bottom fittings or vertically through the glass tube.

The gauge, illustrated in Fig. 154, is designed to prevent accident by scalding to the attendant should the glass tube be broken. A and B are small ball valves which, under working conditions, rest as shown in full lines. But, should the tube burst, the balls will be carried at once into the dotted positions by the rush of steam and water, and held there against the seats by the pressure in the boiler, thus preventing any further escape. The action is practically instantaneous. The cocks D and E are now closed by hand, and F opened, permitting the water to drain off from the gauge, the water in the upper portion escaping through the supplemental tube J. The pressure in the interior of the gauge having been thus reduced to that of the atmosphere, the balls A and B return, by gravitation, to their original positions. A new glass is now inserted, the cocks D and E are opened and F is closed, when the gauge will be again in its normal working order. That proper attention should be paid to the water gauges is of primary importance to secure safe working of a boiler. The gauges should be tested by blowing through several times daily, and no defect should be allowed to pass unremedied. The consequence of a false water level shown in the gauge may be a very serious explosion.

It may be noticed that an empty plain glass tube looks very like the same tube when full of water. Such a tube used for a water gauge, should the water level in it be lost, will fail to indicate whether the water in the boiler is too high or too low. Devices are frequently applied to indicate when the tube is full or empty, such usually taking advantage of the change in appearance of a coloured band at the back of the tube, or of corrugations in a glass plate, caused by the refractive effect of the water.

Shields having glass windows let into a gun-metal frame are often fitted so as to surround the gauge tube; these minimise the risk of accident to the attendant should the tube burst when he is very near the gauge.

Pressure gauge.—Pressure gauges for steam boilers are usually of the **Bourdon type**. The action depends on the tendency which a

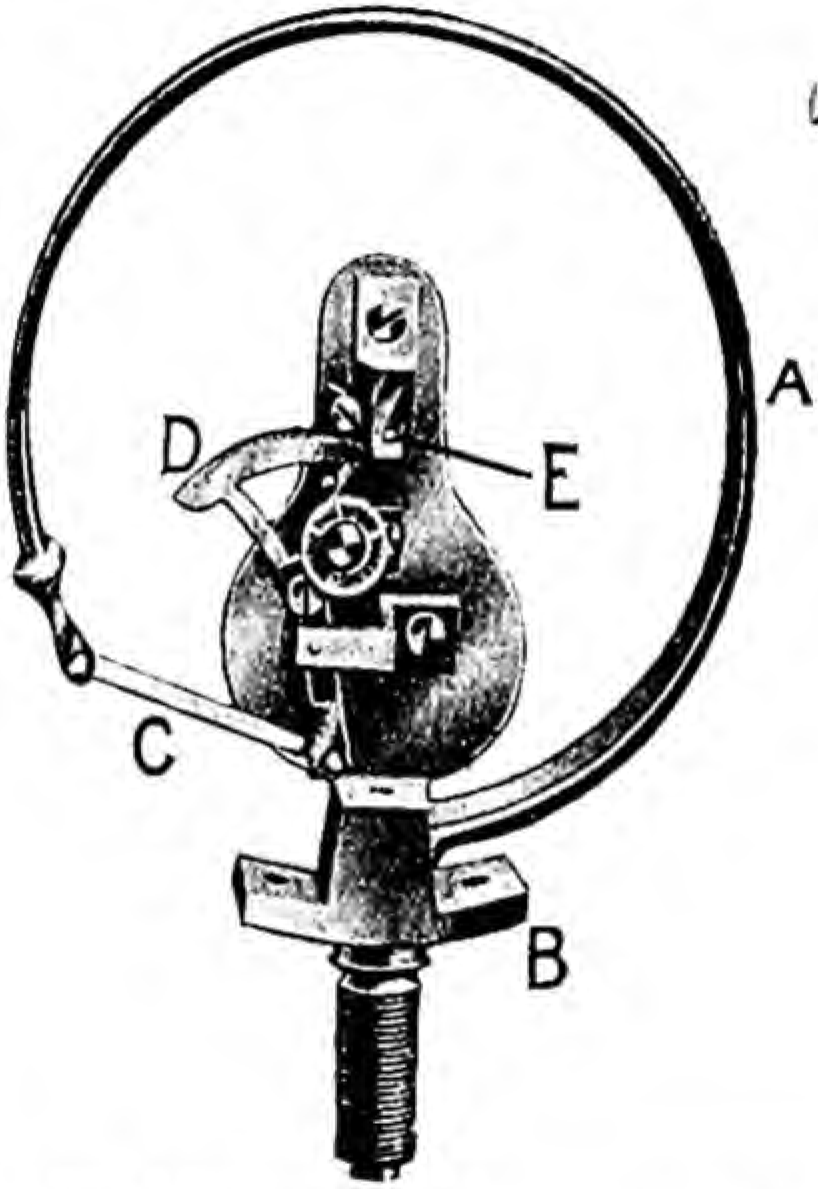


FIG. 155.—Interior parts of a steam pressure gauge.

curved, partially flattened tube has to become straight when subjected to internal presssure.

EXPT. 30.—This effect can be demonstrated easily by attaching a piece of rubber tube about a yard long to a water tap, closing its outer end by a clip, and then bending the tube into a curve lying on the table. On opening the water tap, the rubber tube, which has been slightly flattened by the bending, will distinctly show movement in the attempt to straighten itself.

In Fig. 155 is shown the interior parts of a steam gauge constructed by Messrs. Hopkinson; Fig. 156 illustrates the

exterior. Referring to Fig. 155, A is a flattened tube of hard, solid-drawn phosphor bronze, secured to a bracket B which has passages in it forming the steam inlet to the tube. The free end of the tube is closed, and is connected by means of a short link C to a small toothed sector D. The sector gears with a pinion on the spindle E carrying the outside pointer. The whole of the mechanism is carried by the bracket B. Injury to the mechanism due to straining of the outer case is thus avoided.

Fig. 156.—Steam pressure gauge.

It is bad practice to allow steam into the bent tube, as the elevation of temperature thereby produced is liable to cause injury and false readings. Steam gauges should be sealed off from the direct action of the steam by means of a

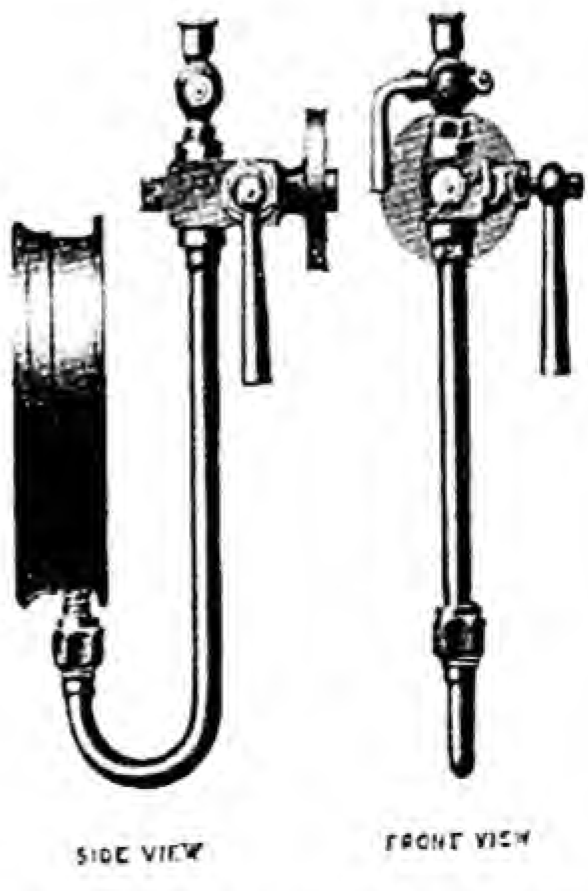


Fig. 157.—Water siphon and tap for steam pressure gauge.

water siphon. This siphon, in its simplest form, consists of a U-tube (Fig. 157); one leg is connected to the gauge and the

other to the boiler, the tube being first filled with water. A tap is usually placed between the U and the boiler.

On opening the tap, the steam pressure will be transmitted to the interior of the flattened tube, causing its free end to move outwards. This movement, transmitted by the mechanism to the pointer, will cause it to rotate over a scale graduated on the outer dial, to an amount proportional to the pressure, and will, therefore, indicate the pressure in the boiler.

Pressure gauges are best graduated by subjecting them to a pressure balanced by a column of mercury. The mercury column gives the actual pressure very accurately, and the scale may be marked with pressures corresponding to its readings. Pressure gauges are liable to alter their readings in the course of time, and should therefore be tested periodically against the readings of a mercury column, or against the readings of an accurate standard test gauge. Excessive heat or cold is to be avoided in actual work, as these cause injury to the tube. Sudden alterations of pressure are very injurious to the elasticity of the tube and should be avoided. Such alterations in pressure may be produced by opening the gauge tap too quickly. In other cases—where the gauge is connected to a steam pipe, or reservoir, where the pressure is varying rapidly—the pointer will take up vibrations. These vibrations can be got rid of by partially closing the gauge tap so as to throttle the opening into the gauge. The best working pressure for a gauge is half that of the maximum graduation, *i.e.* a gauge graduated to 200 lbs. per square inch is suitable for a working pressure of 100 lbs. per square inch.

Vacuum gauges are constructed on the same principle as pressure gauges. The scales are generally graduated to show pressures in inches of mercury below that of the atmosphere. A reading of 5″ on such a gauge means that the pressure in the vessel to which the gauge is connected is 5″ head of mercury less than that of the atmosphere. To obtain the absolute pressure, read the barometer at the same time; supposing this to be 29·5″, the absolute pressure will be $(29{\cdot}5-5)=24{\cdot}5$ inches of mercury.

Lever safety valve.—Fig. 158 shows, partly in section, a safety valve of the lever pattern as constructed by Messrs. Schaffer & Budenberg. *A* is a body secured to a mounting block on the boiler and having a valve *B* resting on a gun-metal

seat. It will be observed that the seat is belled over at the bottom, and therefore cannot come loose. The valve is held down against the steam pressure on its under side by means of a lever

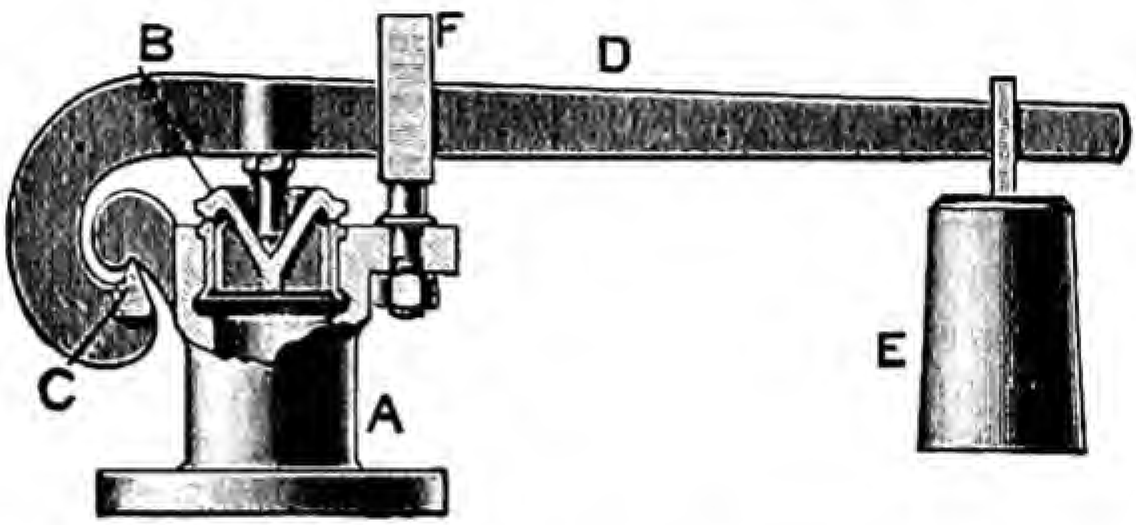


FIG. 158.—Lever safety valve.

D, pivoted on a knife-edge at C, and a weight E, which may be moved to any part of the lever to suit different steam pressures. F is a piece slotted to receive the lever, and permits of a limited upward travel of the lever, thus preventing the valve from being blown off its seat.

The valve will begin to open when the moment of the resultant force acting on it about C is equal to the combined moment of the weight and of the lever about the same point. Thus:

Let p = the steam pressure, lbs. per sq. inch.

d = diameter of the valve, inches.

w = weight of valve, lbs.

W_1 = weight of E, lbs.

W_2 = weight of lever, lbs.

a, b, and c, are respectively the horizontal distances from the pivot to the valve centre, to the centre of gravity of the lever, and to the centre of E (Fig. 159). The centre of gravity of the lever can be found by calculation from the drawing, or, if the lever has been constructed, by placing it on a knife-edge and moving it about

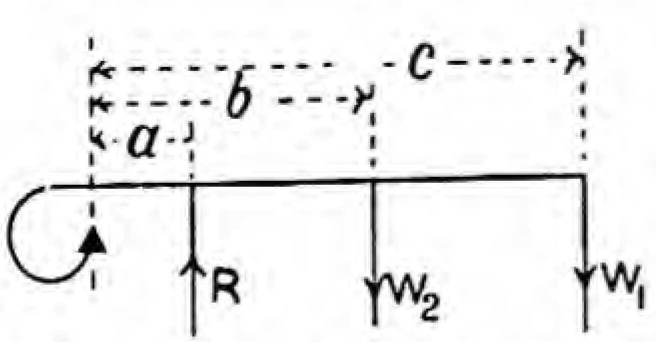


FIG. 159.—Diagram of lever safety valve.

until it balances; the centre of gravity will then be vertically over the knife-edge.

$$\text{Total steam pressure on valve} = p \times \frac{\pi d^2}{4},$$

$$\text{Resultant upward force on valve} = R = \left(p \times \frac{\pi d^2}{4}\right) - w.$$

Taking moments about C,

$$\text{Moment of } R = \text{moment of } W_1 + \text{moment of } W_2,$$

$$Ra = W_1 c + W_2 b,$$

or,

$$\left\{\left(p \times \frac{\pi d^2}{4}\right) - w\right\} a = W_1 c + W_2 b.$$

From this equation, any one of the quantities can be calculated when the others are known.

Dead-weight safety valve.—Fig. 160 shows a safety valve of this type. The valve A rests on a seat B secured to the top of a pipe C, bolted to a mounting block on the top of the boiler. The valve and pipe are covered by a case D, which hangs freely from the valve, to which it is secured by a capped nut. The case D contains weights sufficient to keep the valve on its seat against the normal working steam pressure. Should this pressure become too high, the valve and case will lift, enabling the surplus steam to escape through slots formed in the upper part of the case. The lift of the valve is controlled by the studs E, the heads of which project into wide slots formed in the interior of the case D. It will be noticed that the centre of gravity of the dead load is considerably below the valve. This tends to produce a steady action.

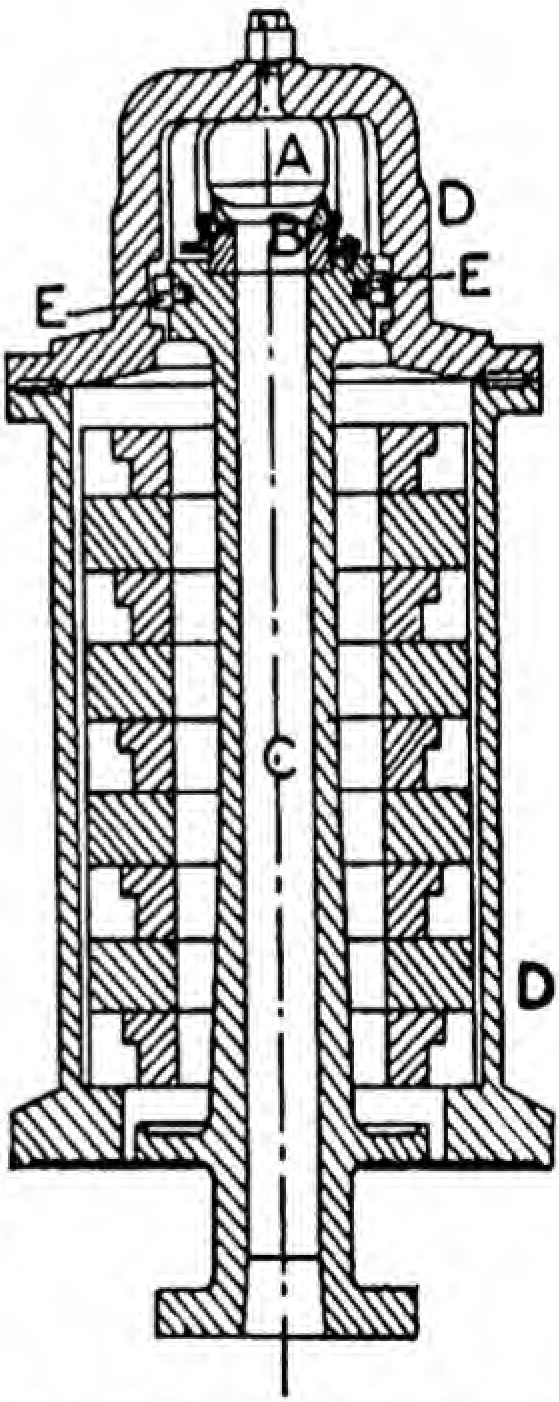


FIG. 160.—Hopkinson's dead-weight safety valve.

Let p = the steam pressure, lbs. per sq. inch.
d = diameter of the valve, inches.
W = total dead load required, lbs.

Total steam pressure on valve $= p \times \frac{\pi d^2}{4}$,

$$\therefore \ W = p \times \frac{\pi d^2}{4} \text{ lbs.}$$

Dead-weight safety valves are very useful, for it is almost impossible for the stoker to overload them. To produce any serious change in the blowing-off pressure, a very large additional weight would have to be applied to the exterior of the case D, and this would be sure to be detected by the inspector.

Generally both a lever and a dead-weight safety valve are fitted to the boiler, and, in order to avoid as far as possible the nuisance of steam discharging into the boiler house from the dead-weight valve, this is set to blow off at a slightly higher pressure than the lever valve. The dead-weight valve thus acts as a check on the lever safety valve.

Low water and high steam safety valve.—Valves of this description are intended to blow off should the steam pressure become too high, or should the water level in the boiler fall to an unsafe extent. Such a valve is shown in section in Fig. 161. The arrangement consists of an outer body A containing the valves, a lower body B to which the valve seat is secured, these bodies being bolted to a mounting block on the top of the boiler. There are two valves, one C being contained within the other D. C rests on a seat formed in the outer valve D, and is held down against the steam pressure by weights E secured to a rod F connected to C. G is a lever pivoted to a support screwed into the mounting block, and having a submerged tile float H suspended from one end and a balance weight K from the other. A knife edge L on the lever is clear of the adjustable collar M in the position shown, but should the water level fall low enough to uncover the float H, the additional pull on the right-hand end of the lever, produced by the loss of the supporting effect of the water on the float, will cause the lever G to move on its pivot. The knife edge L will now engage with the collar on the rod F, which will be pushed upwards, thereby opening the inner valve C and allowing the

steam to escape through passages so constructed as to cause the steam to make a loud noise. The attention of the attendant will thus be drawn to the water level in the boiler. A solid tile float is preferable to a hollow metal one which might be collapsed by the boiler pressure.

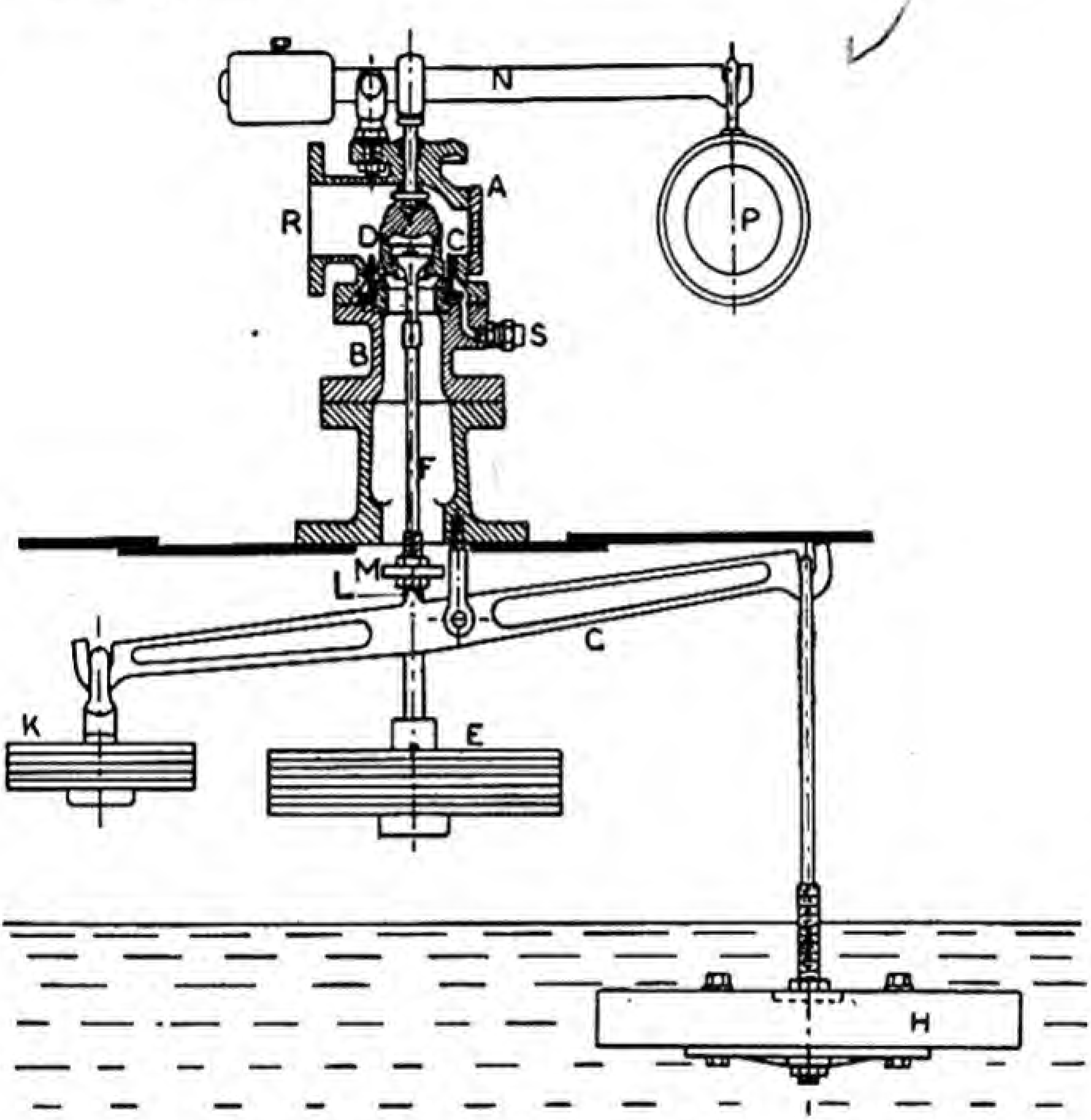


FIG. 161.—Hopkinson's low water and high steam safety valve.

The outer valve *D* is the high pressure steam safety valve. It is held down on its seat partly by the dead weight *E*, and partly by the external lever *N* and weight *P*. In fact, it is a combined dead weight and lever safety valve. Should the steam pressure rise too high, both valves will rise as one, permitting a free escape of steam through the discharge branch *R*. This discharge branch is connected by an escape pipe to the outside of the boiler house so as to discharge the surplus steam into the atmosphere. A small coupling *S* serves for the connection of a pilot pipe discharging

inside the boiler house. Its functions are to drain away water from the valve casing which would otherwise rest on the valve, and also to serve to draw the attention of the stoker, by steam being discharged from it, to the fact that the safety valve is blowing off and that the steam pressure is too high.

Ramsbottom safety valve.—The Ramsbottom type of safety valve is usually fitted to locomotives. It consists of a body secured to the top of the boiler (Fig. 162), and having two vertical branches, *A*, *A*. The open tops of the branches are each closed by a valve *B*. The valves are held down by the pull of a spring *C* placed outside between the branches, secured to the body at the lower end and to a lever *D* at the upper end. The lever has pivots bearing on the valves and is prolonged at one end so as to form a handle by means of which the valves may be tested by hand in order to ascertain that they are not sticking. A downward push on the handle will ease the left-hand valve, which accordingly will be opened by the steam pressure should there be no jamming of the valve on its seat. An upward push applied to the handle will test the right-hand valve similarly. It will be noticed that it is impossible to produce a greater blowing-off pressure by applying a load to the handle. This could only be done by altering the pull of the spring, and this pull is adjusted so as to suit the required pressure.

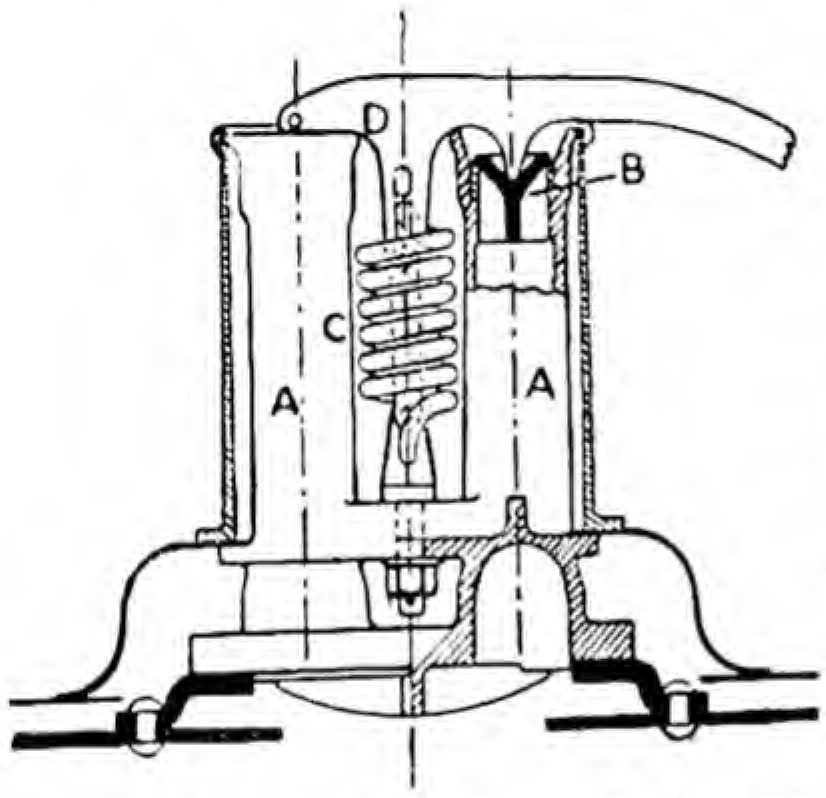


FIG. 162.—Ramsbottom safety valve for G.E.R. locomotive boiler.

The valves and hand lever are prevented from being blown away, should the spring break during working, by means of links which connect the hand lever to the bracket holding the lower end of the spring (Fig. 162). The pin at the upper end of the links works in a slotted hole in the hand lever, thus permitting a working lift to the valves.

Marine safety valve.—Safety valves intended for marine purposes must be of the spring-loaded type. Owing to the movements of the vessel in a sea-way, both lever and dead-weight safety valves are inadmissible. The Board of Trade state, among other instructions, that the safety valves shall be placed directly on the boiler without any intervening pipes; while allowing safety valves under the control of the engineer, there must also be safety valves of the lock-up type, *i.e.* out of the control of the engineer

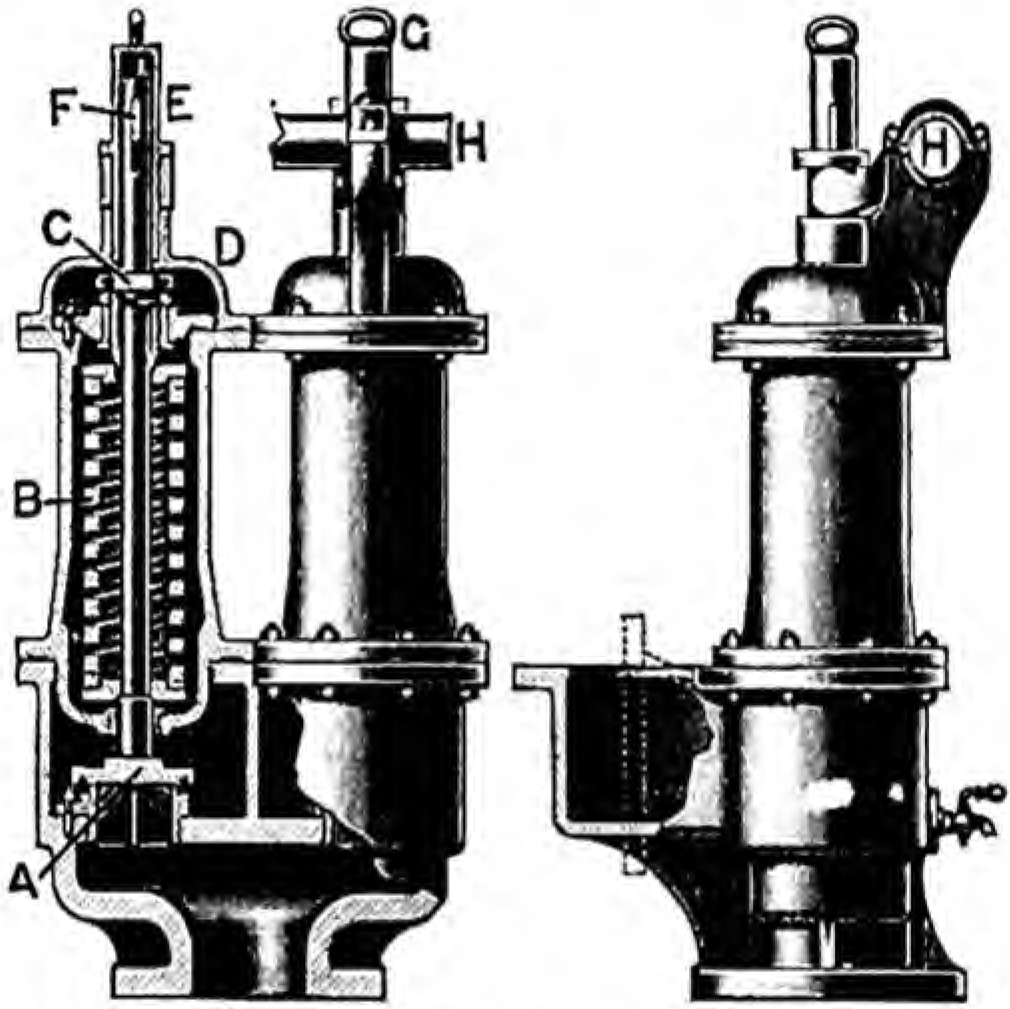


Fig. 163.—Marine safety valve by Messrs. Schaffer & Budenberg.

when steam is up. The object of this is to prevent overloading of the safety valves by the machinery attendants. Lifting gear must be provided by means of which the valves may be eased by hand in order to test that they have perfect freedom to rise.

A pattern of marine safety valve approved by the Board of Trade is shown in Fig. 163. There are two valves *A*, in the same body, each independently loaded by means of a strong spring *B*, bearing directly on the valve spindle. The springs are held down by a gun-metal bush *C*, screwed into a gun-metal liner in the cover. Rotation of the bush enables the pressure of the spring to be adjusted. The valve spindle is connected to the valve, so that if the spindle is raised or rotated, the valve will have a similar

movement. To prevent tampering with the bush--while steam is up—a cover *D* is fitted, held down by bolts. *E* is a sleeve bored to receive the upper end of the valve spindle, and turned to fit the hole in the cover *D*. The sleeve and upper end of the valve

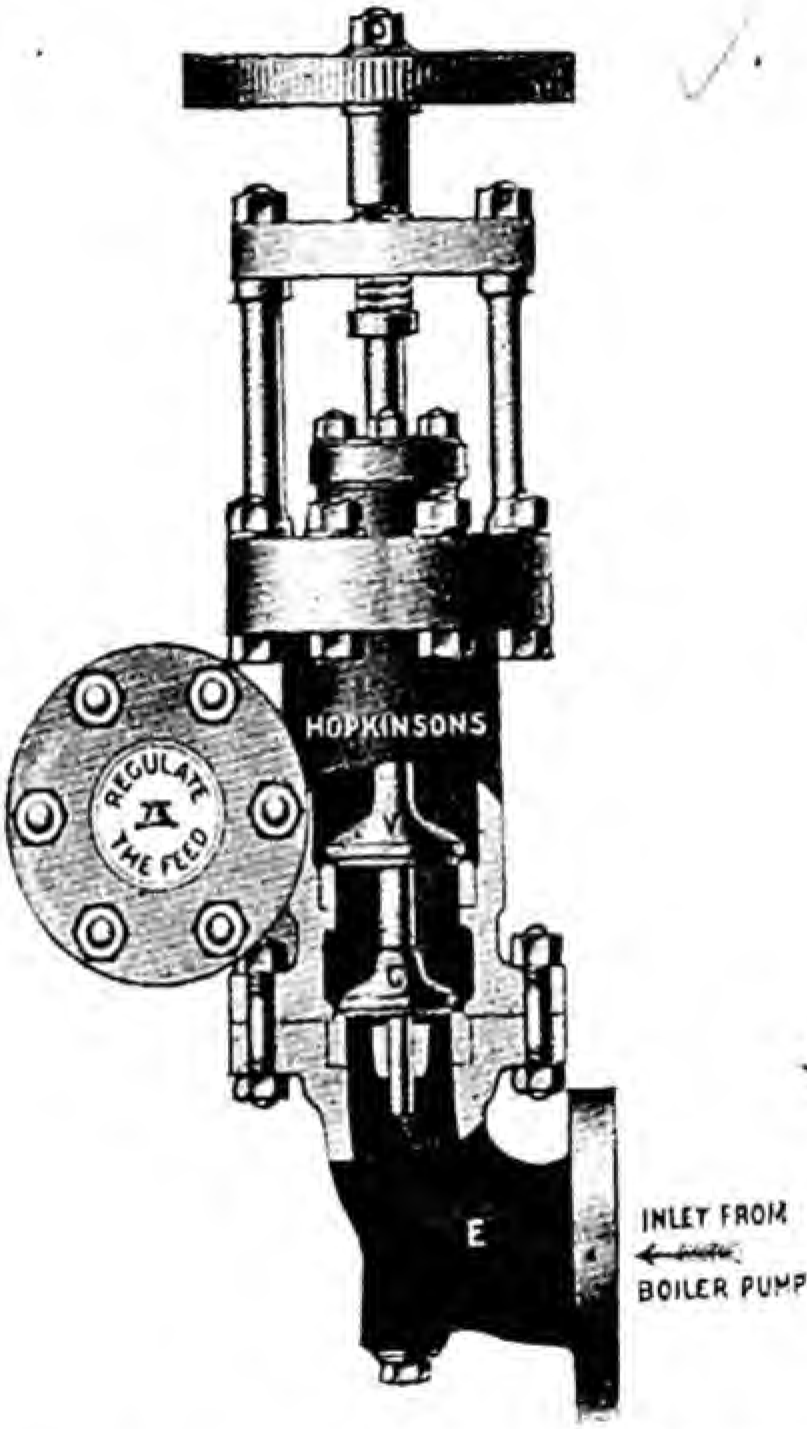


Fig. 164.– Hopkinson's feed regulating and check valve.

spindle are slotted to receive a cotter *F*, the cotter being a fit in the sleeve slots, but the slot in the valve spindle is made longer in order to permit of the steam pressure raising the spindle without thereby raising the sleeve. A bar passed through a hole *G* in the sleeve, may be used to rotate the sleeve and thereby rotate the valve. A shaft *H* has levers fixed to it engaging with collars on the outside of the sleeves. The sleeves, and therefore also the valves, may be raised by rotating this shaft. The shaft is usually connected by levers and rods to a convenient place in the

stoke-hold, whence the safety valves may be eased. It will be observed that it is impossible to overload this valve while steam is up, though, at the same time, there is ample facility for testing.

Feed valve.—To prevent water escaping from the boiler through the feed delivery pipe should the pump cease working, a feed check valve must be fitted. Such a valve is shown in Fig. 164. The feed water travels from the inlet past a check valve *C*—which rises and falls automatically—then past another valve *V*, the lift of which is regulated by means of the hand wheel and screwed spindle, and so into the boiler.

The spindle of the valve *V* is extended downwards, and serves to stop the valve *C* when it has opened sufficiently. The lift of both valves is thus regulated by the same hand wheel.

The arrangement provides for accessibility to the check valve while the boiler is under steam. By closing the valve *V*, the check valve and elbow *E* are isolated from the boiler. The elbow *E* may then be removed by loosening the bolts in the two flanges; the check valve may now be examined, reground to its seat, and replaced.

Marine feed valve.—Fig. 165 shows the feed valve designed for the marine boilers described in Chap. X. *A* is the valve, the lift of which may be regulated by means of the spindle *B* and hand wheel *C*. The spindle is screwed to fit a tapped hole in the bridge *D*. The feed water enters at *E*, passes the valve, and enters the boiler at *F*.

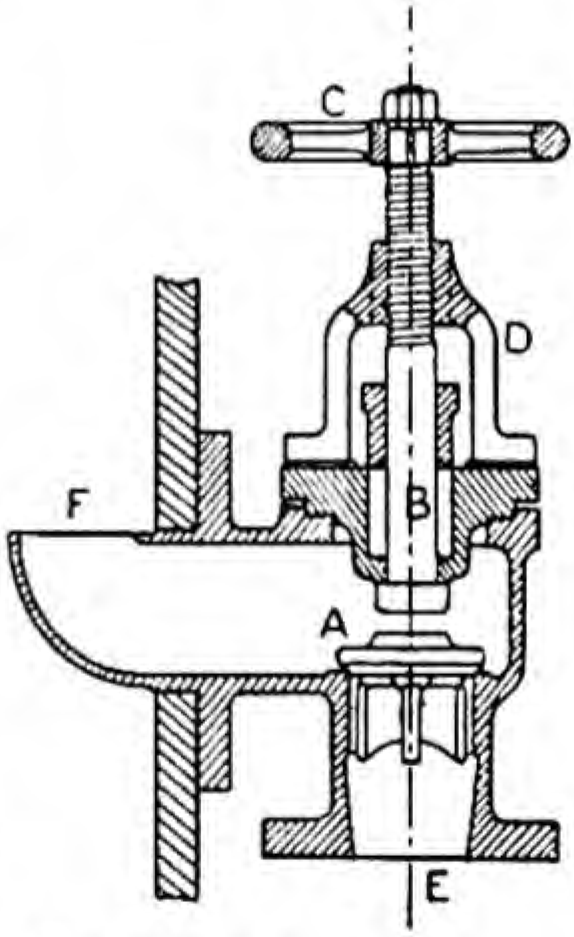


Fig. 165.—Marine feed valve.

Owing to the varying pressure at which the feed water is delivered by the pump, the feed check valve will be continually rising and falling. To prevent breakage, and to minimise wear and tear, the valve is made large in diameter so that sufficient area of passage past it may be obtained with a small lift.

Feed water supply.—Boilers may be fed with water by means of **feed pumps**, which are driven by the main engine (p. 286) or independently by a steam engine or other motor. A well-known feed pump of the latter type is shown in Fig. 166. There are two pumps placed side by side, each pump rod being connected directly to the piston rod of one of the two steam cylinders. In Fig. 166, one pump and its steam cylinder are shown in section. The pump is double acting, *i.e.* it delivers water on each stroke forward and backward, there being suction and delivery valves for each side of

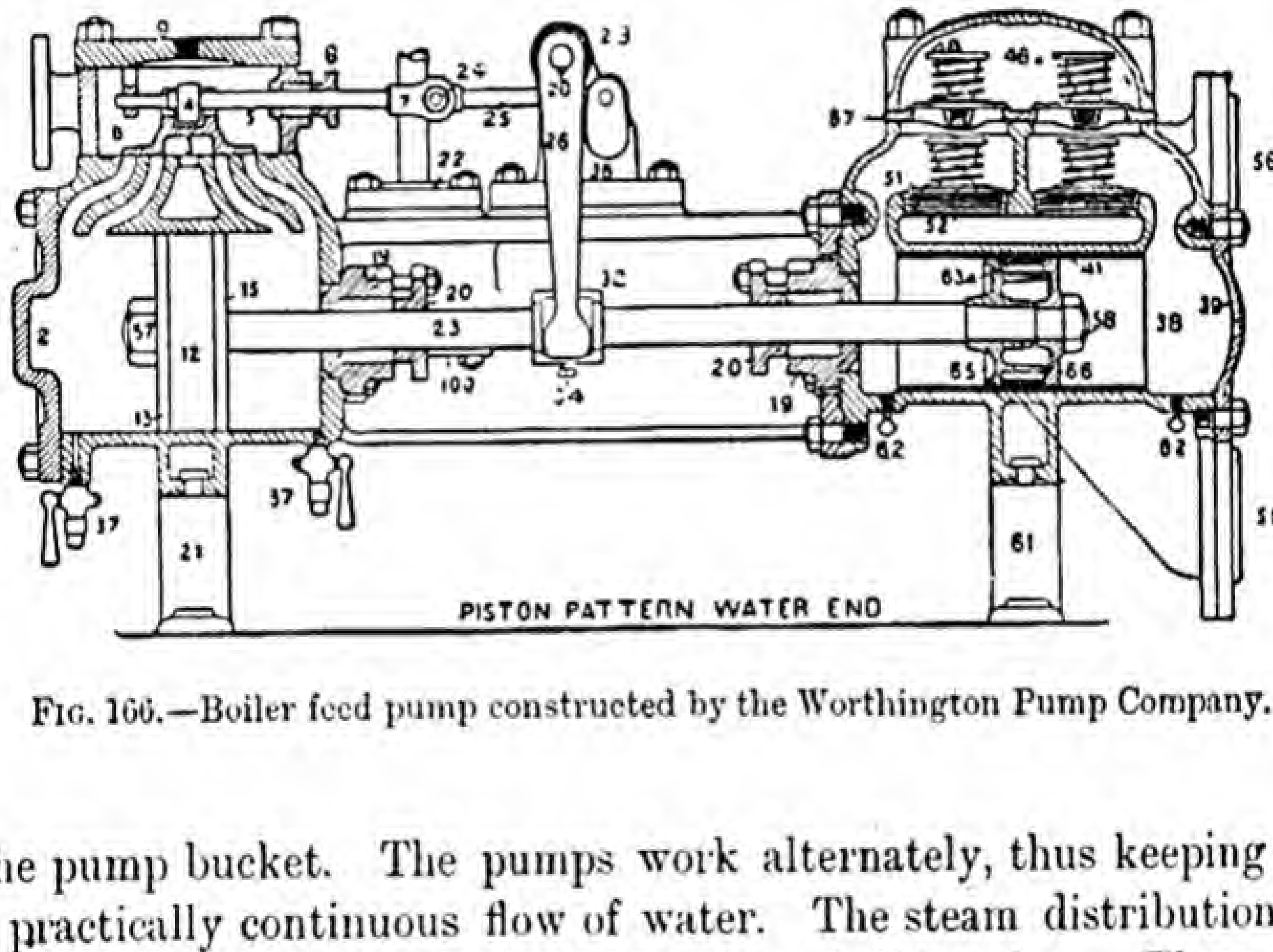


FIG. 166.—Boiler feed pump constructed by the Worthington Pump Company.

the pump bucket. The pumps work alternately, thus keeping up a practically continuous flow of water. The steam distribution to each cylinder is accomplished by means of slide valves. There are no rotating parts in the appliance, the slide valve of one cylinder being operated by means of a lever connection to the crosshead of the other cylinder.

The injector.—Another means of supplying feed water is by means of an **injector**. In this appliance there are no working pistons or plungers, the water being fed into the boiler by the action of steam flowing through a tapering nozzle. Fig. 167 shows in section an injector made by Messrs. Holden & Brooke, Ltd. Steam enters the injector through S and passes into a passage or **steam cone** E. Water enters the injector at W and mingles with the steam at the **combining cone** K. Condensation of the steam ensues, thus producing a partial vacuum which maintains an

inward rush of water. The passage from *K* to *D′* converges, hence *D′* will be the place where the velocity of flow attains its maximum value. The passage then diverges, producing a reduced velocity towards *L*, and thus, by the well-known law of fluids, gives an increased pressure. The dimensions are so arranged that the pressure produced at *L* is greater than that in the boiler, thus enabling the mingled feed water and water of condensation from the steam to flow through the delivery pipe into the boiler.

The steam cone is adjusted by means of a screwed spindle *F*, which may rotate, but is rendered incapable of axial movement by a collar *G*. Rotation of the handle *A* will thus cause the steam cone *E* to rise or fall in the injector body, and so, at the same time, adjust both the steam opening at *S* and the water opening at *K*. A scale of steam pressures at *B* enables the adjustment to be made rapidly by bringing the pointer *A* opposite the scale pressure corresponding to that in the boiler. Surplus water is got rid of through gaps at *D*, *D′*. In working, the injector is adjusted so that no water escapes through these gaps. A valve is fitted at *M* to enable the injector to restart without attention after any temporary failure in the steam or water supply. The valve opens outwards, permitting a free outlet for steam or water, but closes directly the vacuum is recovered by the normal conditions being restored.

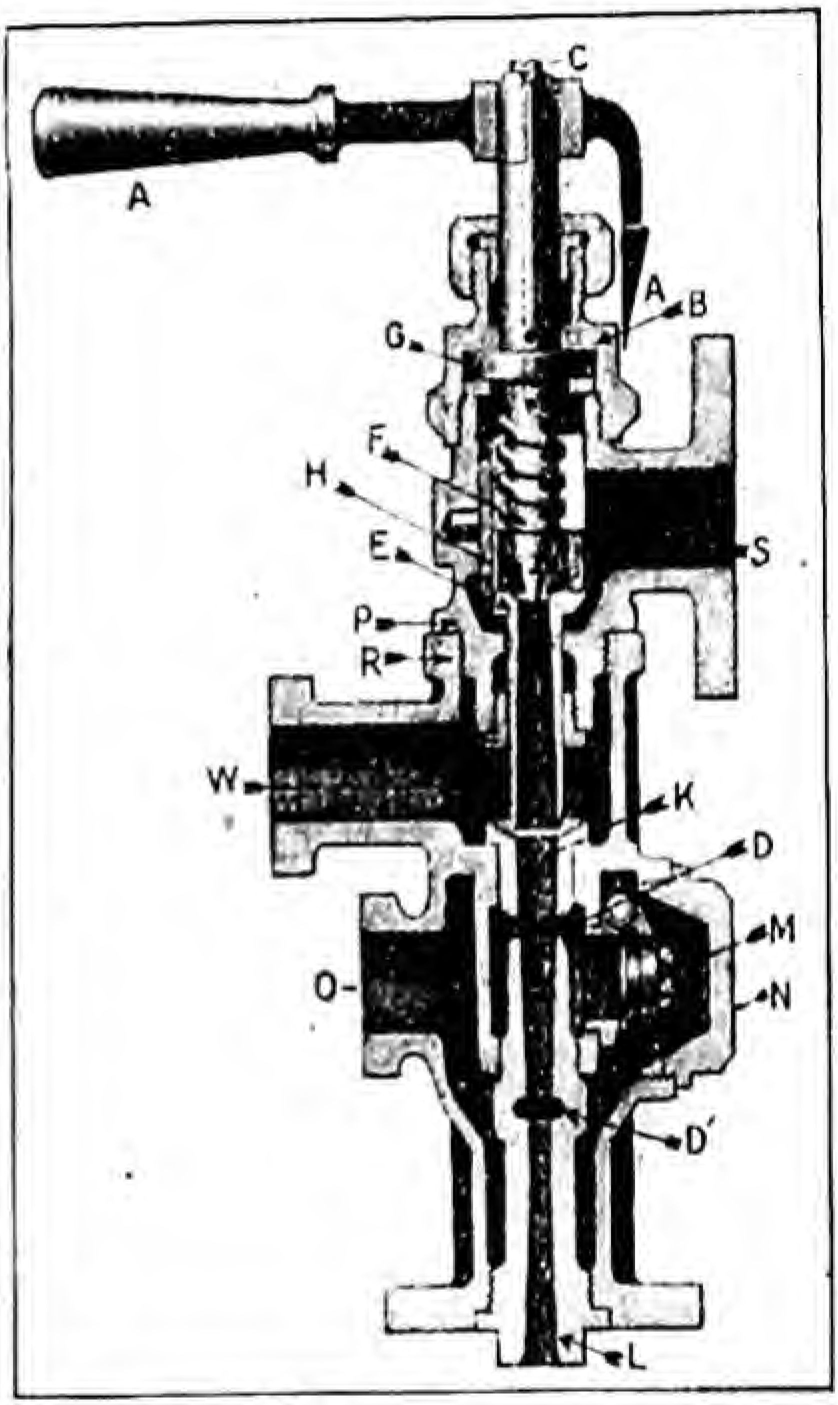


Fig. 167.—Section of an injector.

Blow-off valve.—The purposes to be fulfilled by the **blow-off**

valve are to empty the boiler of water when required for periodical inspection of the interior, and also to enable the stoker daily to blow off some of the water in order to sweep out some of the mud which settles on the bottom of the boiler.

The example shown in section in Fig. 168 consists of two short cylindrical valves *A* and *B*, sliding one within the other and forced apart by a short spring placed inside. The ends of the cylinders bear on steel seats, *C* and *D*, screwed to the body, thus closing the passage. The passage is opened by the valves *A* and *B* being slid into the left-hand portion of the body; this sliding is effected by a rack *E* gearing with a pinion *F* forming part of a spindle *G*, which may be operated by a large double-handed box-spanner pushed through a hole in the front foot-plate of the boiler (Fig. 121).

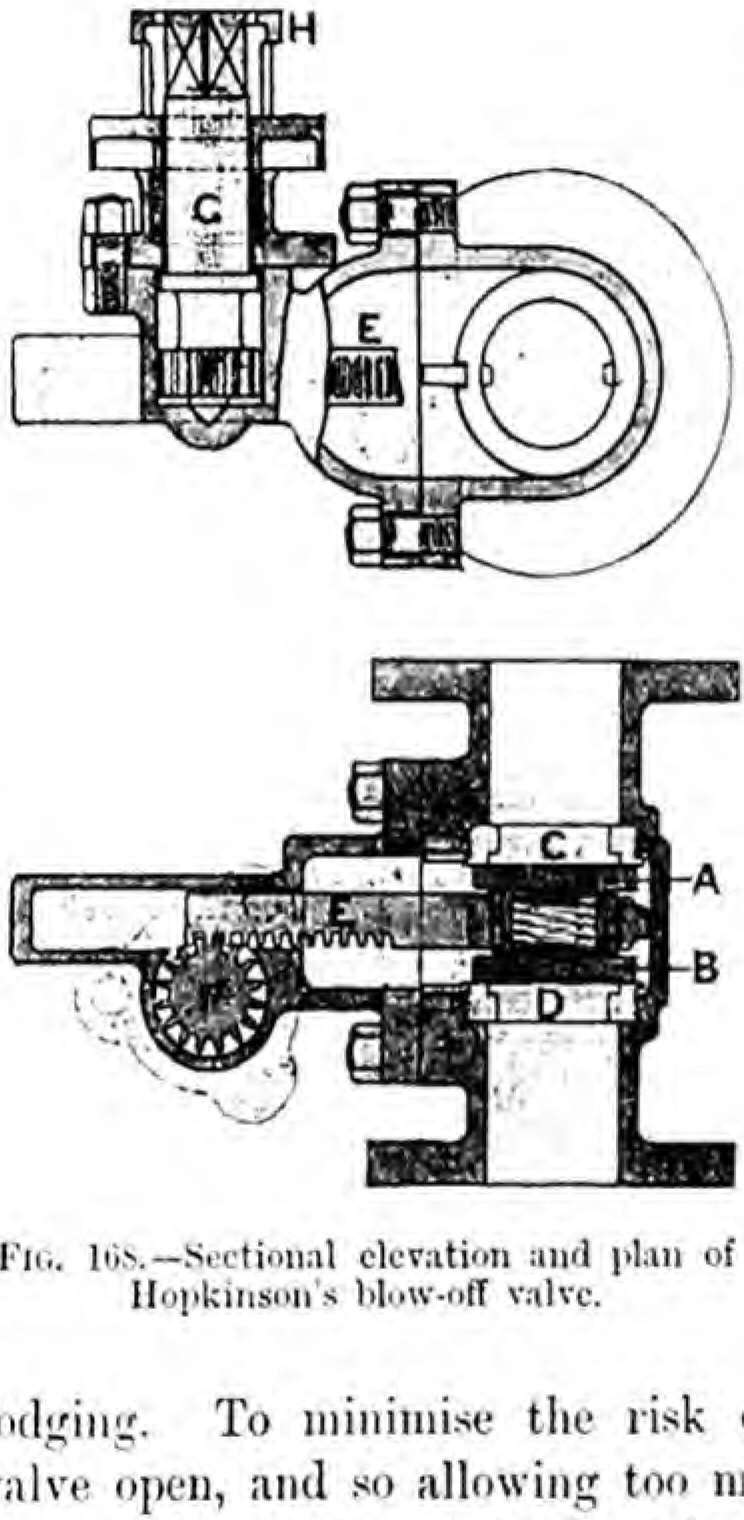


FIG. 168.—Sectional elevation and plan of Hopkinson's blow-off valve.

The valves, being slid to one side, leave a straight passage through the body, thus enabling the water to sweep mud, etc., through the valve without any of it lodging. To minimise the risk of the attendant leaving the valve open, and so allowing too much water to escape, the box-spanner has a snug formed on its end. To cause the spanner to engage with the square formed on the end of the spindle *G*, the attendant must pass the snug on the spanner through a slot in a cap *H* enclosing the square. On the spanner being turned to open the valve, the snug will prevent the spanner being withdrawn until the valve is closed again, when the snug will again come opposite the slot. As the spanner, projecting above the front foot-plate, forms an obstacle to the free movements of the

attendant, it is not likely to be left in position longer than can be avoided.

It is very important that blow-off valves should not permit any leakage from the boiler, and frequent examination should be made to ensure this precaution.

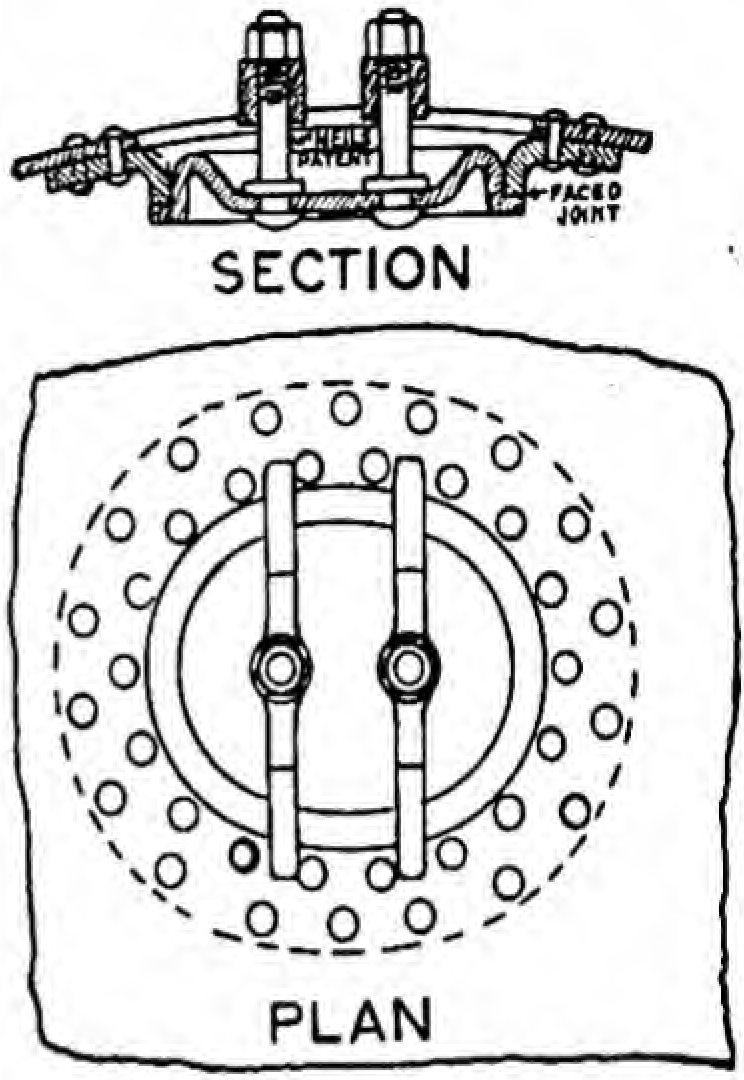


FIG. 169.—M'Neil's door.

Manhole doors.—M'Neil's door, for covering manholes and mudholes, is used largely. The door is embossed by hydraulic pressure from a steel plate (Fig. 169). A flanged compensating ring or saddle, made of steel, is curved to fit the boiler plate and is machined flat so as to form a faced joint with the door. The door is placed inside the boiler, and is of oval shape in order that it may pass through the hole in the saddle. The door is held in position by two studs, with nuts screwed against two external bridges bearing on the edges of the hole. The steam pressure, acting on the interior, pushes the door outwards and so assists in keeping a tight joint without putting undue forces on the studs.

FIG. 170.—Fusible plug, by Schaffer & Budenberg.

Fusible plug. — This device is intended to operate, should the water level in the boiler fall too low, by permitting a discharge of steam into the furnace, thus quenching the fire. It consists of a gun-metal body, Fig 170, screwed into the top of the

furnace tube over the grate, and having a hole closed by a small plug held in its place by easily fusible metal. Should the plug be uncovered by reason of the water level falling too low, the fusible metal is melted by the heat of the furnace; the plug is then blown out by the steam pressure, and a rush of steam into the furnace takes place. To be at all trustworthy, fusible plugs should be scraped clean very frequently, and yearly should be renewed entirely.

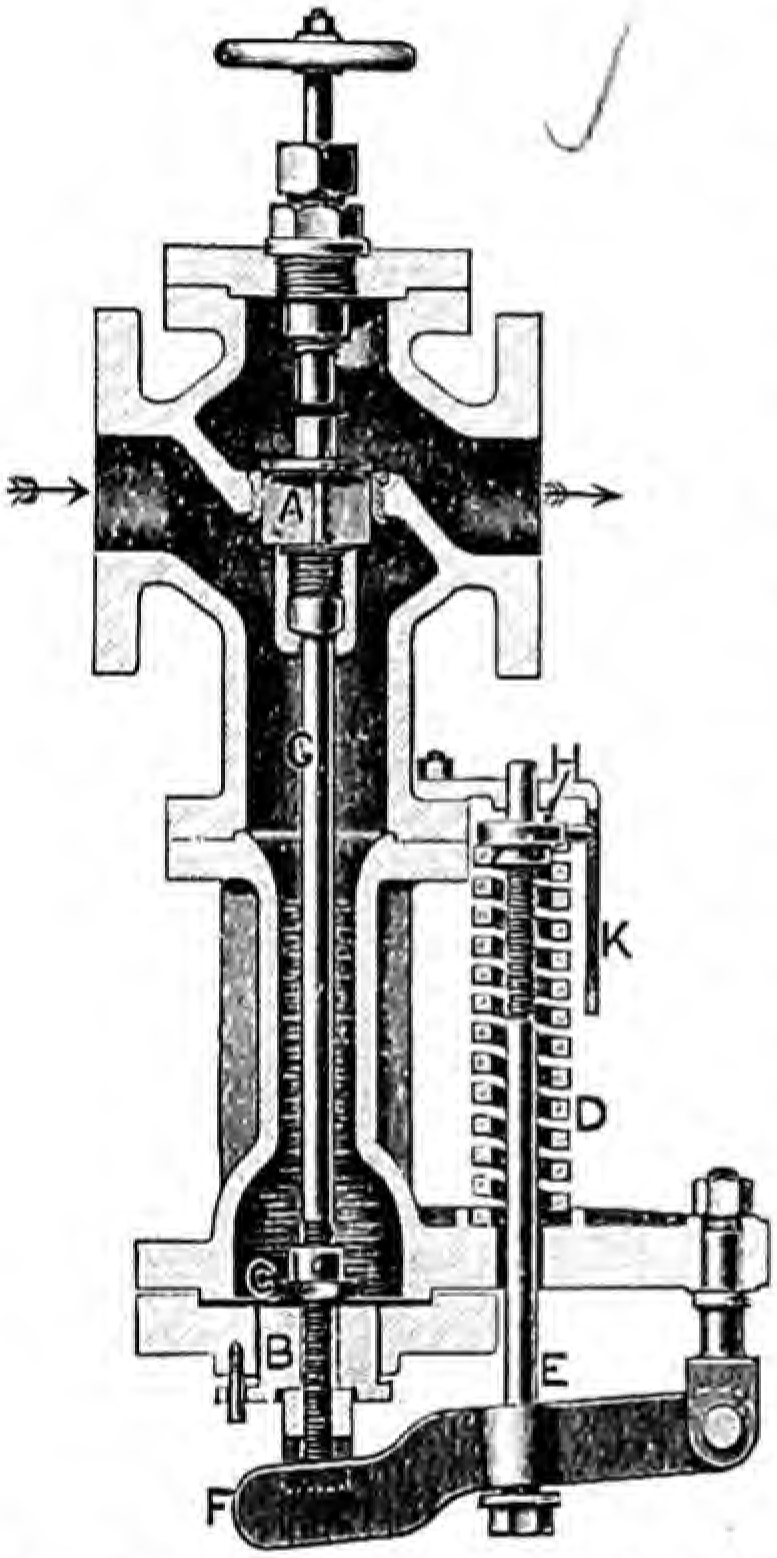


Fig. 171.—Sectional view of a reducing valve.

Reducing valve.—Steam is frequently required for various purposes at a lower pressure than that at which it is generated in the boiler. To enable this reduction of pressure to be effected a **reducing valve** is employed, the function of which is to "wiredraw" the steam and so lower it to a steady predetermined pressure.

In Fig. 171 is shown in section a reducing valve of the Auld type as made by Messrs. Schaffer & Budenberg. In this valve the steam enters at the top left-hand side, is throttled by the poppet valve A, and is discharged at a lower pressure on the right-hand side. At the bottom of the casing is a plunger B and a rubber diaphragm C. The steam pressure acts on these, pushing them downward against the resistance of a spring D, communicated by a rod E and lever F. The plunger is connected to the valve A by a rod G. The areas of the plunger B and of the valve A are such that the upward steam pressure on A is balanced by the downward pressure on B. An additional downward force is obtained from

the reduced steam pressure acting on the top side of the valve *A*, and this is balanced by the spring *D*. Any excess pressure on the top side of the valve *A* will result in the valve being pushed downwards, and thus cause further throttling of the steam until the pressure on the top side of *A* falls again to the required value. The spring *D* may be adjusted to give varying reduced steam pressures by rotating the rod *E* by means of the hexagonal head at its lower end. This rotation causes the nut *H* further to compress or release the spring. *K* is a scale, showing, by the position of a pointer on *H*, the pressure to which the valve is adjusted to reduce. The water of condensation lying in the lower part of the casing serves to protect the rubber diaphragm from the direct heat of the steam.

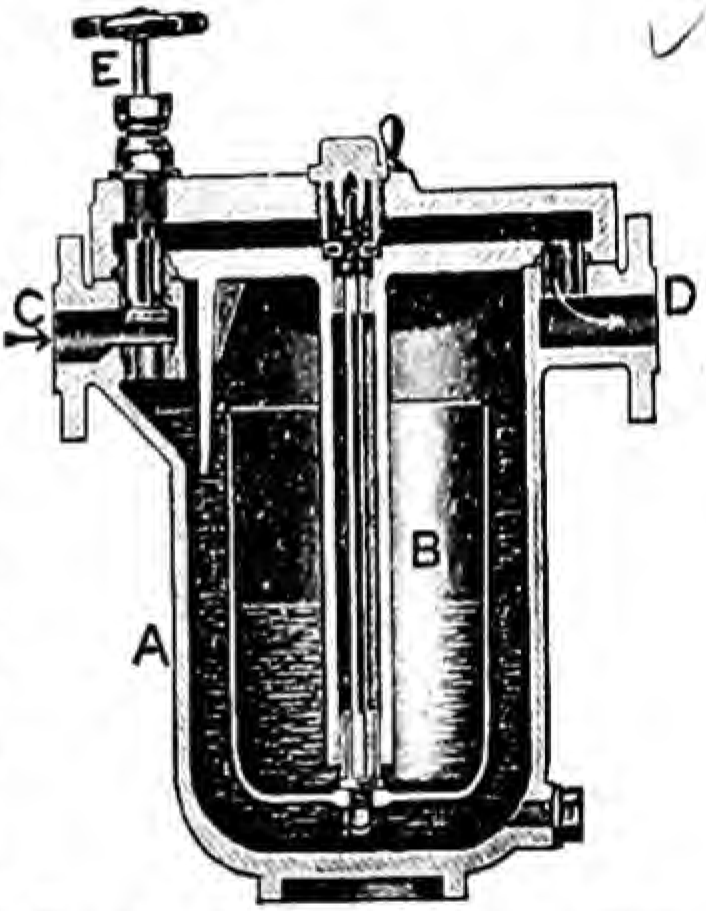


FIG. 172.—Section of a float steam trap.

Steam traps.—Steam traps are devices intended to drain off water accumulating in the steam pipes while, at the same time, the escape of steam is prevented. There are many different types, some of which are here described.

Float steam trap.—Steam traps of this type are intended to operate by a float which is raised when water accumulates in the trap, thus opening a valve and permitting the water to escape. In Fig. 172 is shown in section such a trap manufactured by Messrs. Schaffer and Budenberg. The trap consists of an outer closed vessel *A* fitted with a screw-down lid on the top. *B* is the float, open at the top, and is guided vertically by means of a central rod to which also the discharge valve is connected. *C* is the inlet and *D* the outlet. A valve *E* is provided, to be operated by hand, and permits blowing through the by-pass (shown black under the cover), thus permitting large accumulations of water to be got rid of. It is intended that the steam pressure on the surface of the water in the float should be sufficient to force it up the central tube and so discharge it.

Assuming that the float is empty and that there is water in the outer casing, the float will be elevated, and the discharge valve will be closed. As more water collects, its level will rise until it begins to overflow into the float. Presently the weight of water in the float will be sufficient to cause it to descend, opening the discharge valve as it does so, and thus enabling the steam pressure to discharge the water in the float. The water in the outer vessel will not be so discharged as it is sealed off from the discharge orifice by the float. As the weight of water is reduced in the float, it will rise again and so close the discharge valve. Thus, water alone will be discharged from the trap. This trap must, of course, be arranged vertically as shown in the illustration.

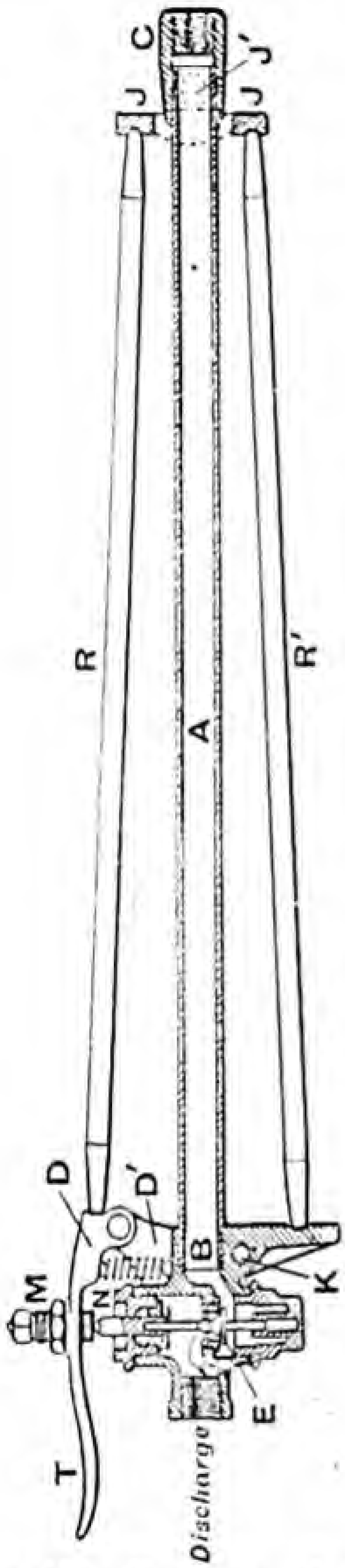


FIG. 173.—Expansion steam trap.

Expansion steam trap.—In expansion steam traps advantage is taken of the expansion of materials on heating, the movement thereby obtained being utilised to operate a valve. Fig. 173 shows in section such a trap constructed by Messrs. Holden & Brooke, Ltd. The pipe *A* is connected at *C* to the part to be drained of water; its other end is fitted with a valve *E* closing the discharge outlet. This valve is operated, in a manner to be described, by the lever *D* and the rods *R* and *R'*. *J*, *J* is a small bracket which may rock on a pivot *J'* secured to the end of the pipe *A*. The rod *R'* bears at one end against the valve bracket, and at the other end against *J*. The rod *R* bears against *J* at one end and against the lever *D* at the other end. The pipe *A* will be heated and expand in the direction of its length should steam be admitted to it, and will contract if it contains water colder than the steam. The rods *R R'* are practically out of the reach of heating and cooling action, and so remain of constant length.

Steam being admitted to the tube *A*, it becomes of greater length, and as the rods *R R'* remain of the same length as at first, the lever *D* will move outwards (assisted by the spring *D'*), thus enabling the steam pressure in *A* to close the valve *E*. An accumulation of water in *A* will cause it to cool and contract. The rods *R R'* will then operate on the lever *D* and thus open the valve *E*, permitting the water in *A* to escape. *D* is extended at *T* to form a handle; by depressing this handle the valve may be opened for blowing through. *M* is an adjusting piece for regulating the valve opening.

In the Geipel trap, the different amounts of expansion of a brass and an iron tube, caused by steam flowing through them, is utilised for operating the discharge valve.

Steam separators.—The object of a steam separator is to remove, as far as possible, fine particles of water carried along with the steam on its way from the boiler to the engine. Water may be present in the steam due to priming in the boiler, but even if this has been removed there will always be a certain amount of condensation in steam pipes along which saturated steam is flowing, and this water, if allowed to enter the cylinder, will induce further condensation during the admission period. There are many different forms of **separators**, that shown in Fig. 174 being manufactured by Messrs. Holden & Brooke, Ltd. In this separator, the steam is compelled to whirl and also to reverse its direction of motion

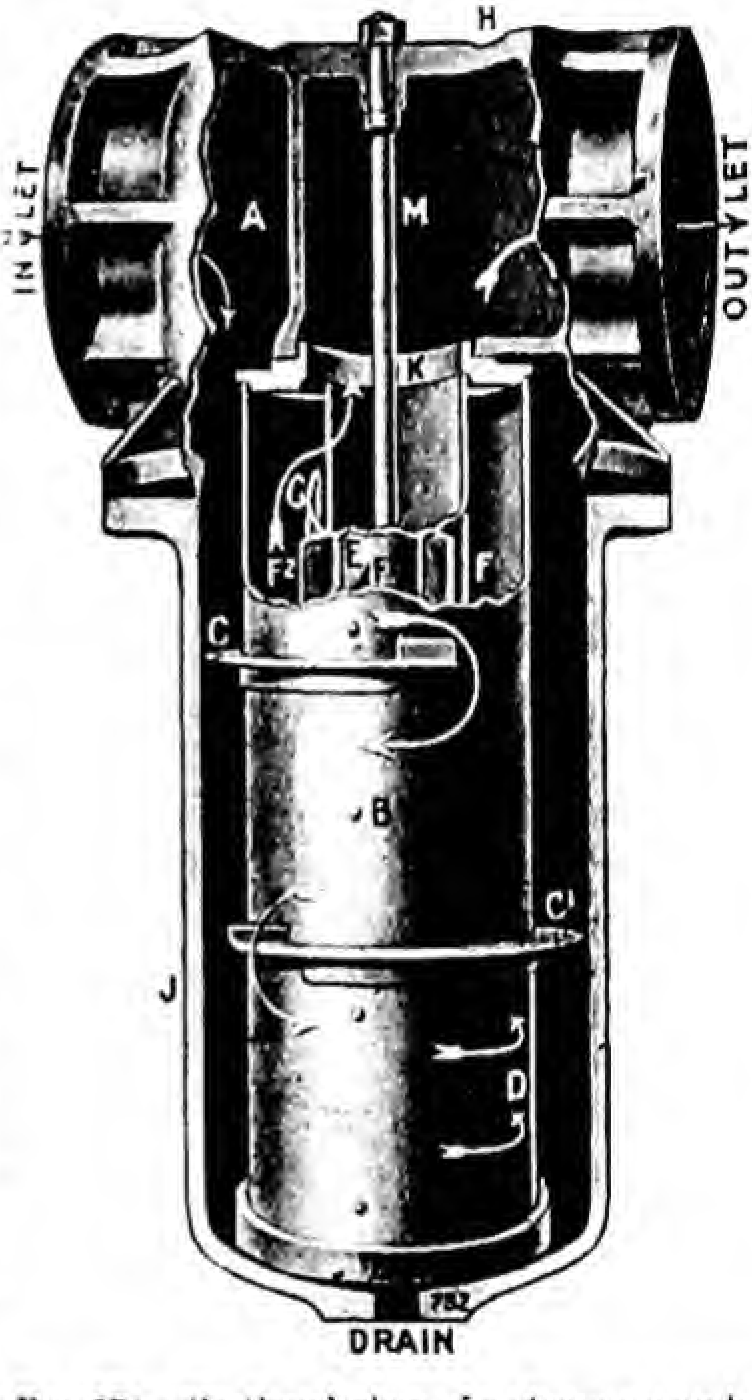


FIG. 174.—Sectional view of a steam separator.

several times, while its velocity of flow in the separator is lower than that in the steam main.

Steam enters the separator at *A*, and flows downwards between the casings *B* and *J*. *C* and C^1 are galleries secured to *B*, and to pass these, the steam has to travel in a circular direction and also twice to reverse its direction of motion. The steam particles do this with comparative ease, but the larger inertia of the water particles causes them, in the endeavour to preserve straight line motion, to be thrown against the inner walls of *J*, whence they find their way to the drain at the bottom. The steam, having reached the bottom of *J*, passes through a port *D* in *B* into a space between *B* and a third cylinder *E*; having travelled upwards through this space it is allowed to pass through *K* and *M* into the outlet main. There are vertical projections *F*, F^1, F^2, secured alternately to the outer wall of *E* and the inner wall of *B*. These cause the steam passing between *B* and *E* to take a sinuous course and aid in the removal of any water particles which may have escaped being discharged in the outer chamber.

EXERCISES ON CHAPTER XI.

1. Give sketches and a description of any form of junction or main stop valve, or a locomotive regulator.

2. A stop valve is 8″ diameter, the working pressure being 120 lbs. per square inch. Calculate the total pressure on the valve when closed. Suppose the stress on the valve spindle to be limited to 5000 lbs. per square inch, find the sectional area and the diameter of the spindle.

3. Give sketches and description of any form of water-gauge.

4. Explain the action and give sketches showing the construction of a pressure or vacuum gauge.

5. Sketch and describe the construction of any pattern of safety valve. In the valve you select, explain any way in which it is possible for the attendant to alter the setting of the valve so as to cause it to blow off at an increased pressure.

6. The diameter of a safety valve of the lever pattern is 3½″. The weight of the lever is 10 lbs. and the centre of gravity is 15″ from the fulcrum. The movable weight on the lever is 90 lbs. The weight of the valve is 3½ lbs. and the distance from the fulcrum to the centre of the valve is 4¾″. Calculate the distance of the weight on the lever from the fulcrum in order that the valve may blow off at a steam pressure of 80 lbs. per square inch.

7. Sketch and explain the construction and working of a feed check valve.

8. Explain the construction and working of an injector. Give sketches.

9. Sketch and describe any pattern of blow-off valve. What precautions must be observed in this form of valve?

10. What is the function of a reducing valve? Give sketches and description of any type of reducing valve for steam.

11. Explain the working and give sketches of any kind of steam trap.

12. What is the object in fitting a separator to a steam pipe? Sketch and describe any type of separator.

13. Sketch the construction of a lever safety valve with balance weight, and state under what circumstances such a construction could not be used. If the lever be 16 inches in length and the centre of the valve seat is 4 inches from the fulcrum, while the diameter of the valve is 4 inches, find the weight to be placed at the end of the lever so that steam may blow off at a pressure of 45 lbs. per square inch, the weight of the valve and of the lever being neglected. 1896.

14. Sketch and describe the construction and action of a non-return feed-water valve for either a land or a marine boiler. Where and at what level is such a valve placed on the boiler? 1896.

15. Describe and show by a sketch the construction of Ramsbottom's safety valve for a locomotive engine. How are the lever and valves prevented from flying off in the event of the spring breaking? If in a Ramsbottom valve the two valves each have a diameter of $2\frac{1}{4}$ inches, what would be the pull on the spring when the steam is just blowing off at a gauge pressure of 140 lbs. to the square inch (neglect the weight of the valves and connections). 1897.

per square inch.

CHAPTER XII.

STRENGTH OF BOILERS.

Stress.—The material of a body is said to be under **stress** when, if a given section be taken, forces are communicated by the portion of the body which is situated on one side of the section to the portion situated on the other side. **Stress is measured by stating the magnitude of the force per unit area of the section**; thus:

$$\text{Stress} = \frac{\text{total force}}{\text{area of section}}.$$

The units will be pounds or tons per square inch depending on the units employed in stating the force, the area being stated in square inches.

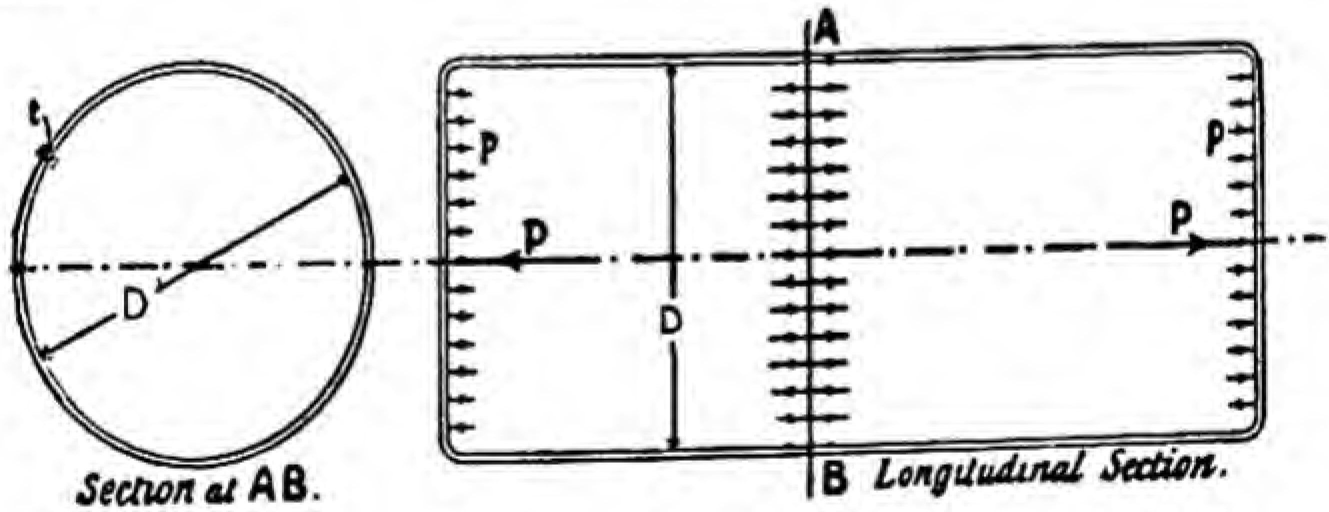


FIG. 175.—Stresses on a circumferential seam of a cylindrical shell.

If the forces are of the nature of **pull**, the stress produced is described as **tensile**; **push** forces produce **compressive stress**; should the forces tend to make one portion of the body slide on the other portion, the section is said to be under **shear** stress.

Stresses in a cylindrical shell.—Consider a closed cylindrical vessel with walls of small thickness compared with the diameter.

On subjecting the vessel to internal fluid pressure, stresses will occur in the material. Taking any circumferential section, not too near the ends of the shell, there will be tensile stress uniformly distributed over it, due to the pressures on the end plates. Referring to Fig. 175.

Let D = internal diameter of shell, inches;

p = fluid pressure, lbs. per square inch;

t = thickness of the plate, inches;

P = total pressure on the end of the shell; then

$$P = p \times \frac{\pi D^2}{4}, \text{ lbs.}$$

The area of the circumferential section of the shell is given approximately by

$$\text{Sectional area} = \pi D \times t, \text{ square inches};$$

$$\therefore \text{ stress on section } AB = \frac{P}{\pi D t}$$

$$= \frac{p \times \frac{\pi D^2}{4}}{\pi D t}$$

$$= \frac{pD}{4t} \text{ lbs. per square inch.}$$

The stress on a longitudinal section of the shell must now be determined. The whole length of the shell need not be considered, but only the portion between two cross sections taken one inch apart; for if this portion is taken sufficiently far from the ends of the shell, the **staying action** of the ends becomes negligible, and the stresses on all such rings will be alike. The fluid pressure on the ring is represented by the arrows shown (Fig. 176), everywhere directed perpendicular to the curved surface of the ring. Take components of these, as shown,

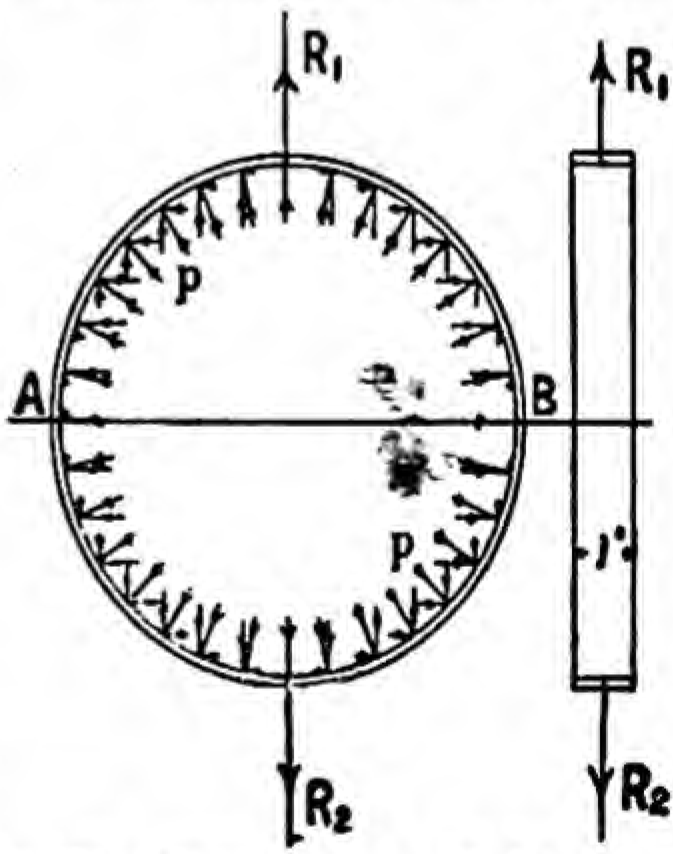


FIG. 176.—Ring cut from a cylindrical shell.

parallel and perpendicular to the diameter AB, and consider the portion of the ring above AB. The horizontal components acting towards the right and left will balance one another, and need not be further considered. The vertical components will have a resultant R_1. Similarly, on the portion below AB, a resultant force R_2 will act, equal and opposite to R_1. These two forces put the material of the ring at A and B under tensile stress, which must now be calculated.

First to obtain R. There will be no difference experienced in the equilibrium of the ring if we imagine it to be filled up to the

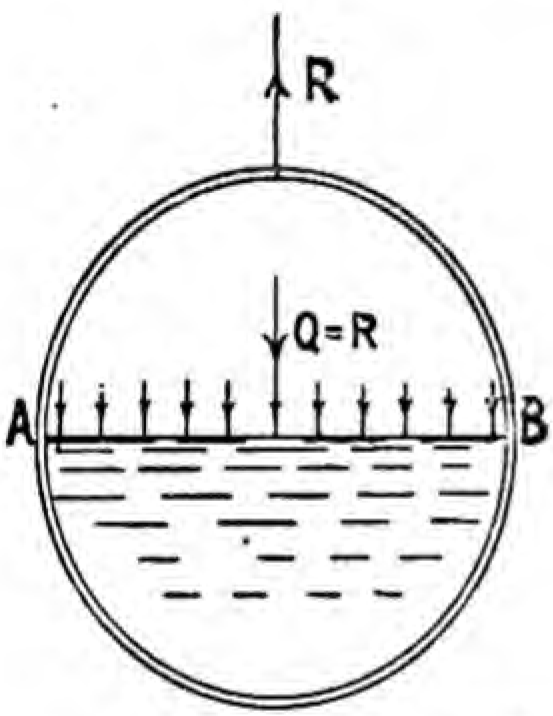


FIG. 177.—Diagram showing the magnitude of R.

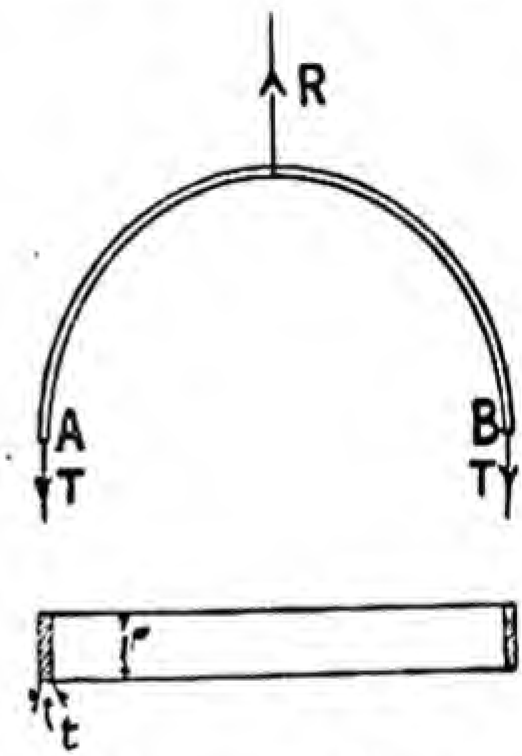


FIG. 178.—Stresses on a longitudinal seam.

level of AB with cement. The pressure on the surface of the cement will be perpendicular to AB (Fig. 177) and the resultant force due to this will be

$$
\begin{aligned}
Q &= p \times \text{area of surface of } AB \\
&= p \times D \times 1.
\end{aligned}
$$

R and Q now preserve the equilibrium of the ring, therefore

$$
\begin{aligned}
R &= Q \\
&= p \times D.
\end{aligned}
$$

Imagine the material at A and B to be cut, and that forces T, T are supplied at the sections in order to balance R (Fig. 178); then if the stress on the sections at A and B be called q,

$$
\begin{aligned}
T &= q \times \text{area cut at } A \text{ or } B \\
&= q \times t \times 1.
\end{aligned}
$$

Also $$R = 2T.$$

$$\therefore \quad pD = 2 \times q \times t;$$

$$\therefore \quad q = \frac{pD}{2t} \text{ lbs. per square inch.}$$

It has already been found that a circumferential seam has a tensile stress equal to $\frac{pD}{4t}$, consequently **the stress on a longitudinal seam is just double that on a circumferential seam.** For this reason **boilers** are constructed with the longitudinal joints much stronger than the circumferential joints.

EXAMPLE. A boiler shell is 6 feet in diameter, and the metal is $\frac{1}{2}''$ thick. If the steam pressure is 100 pounds per square inch, calculate the stress on circumferential and longitudinal sections.

$$\text{Total pressure on end of shell} = 100 \times \frac{\pi D^2}{4};$$

$$\text{Area of circumferential section} = \pi D t;$$

$$\therefore \text{ Stress on circumferential section} = \frac{100 \times \pi D^2}{4 \pi D t} = \frac{100 \times D}{4 \times t}$$

$$= \frac{100 \times 72}{4 \times \frac{1}{2}}$$

$$= \underline{3600} \text{ pounds per square inch.}$$

$$\text{Stress on longitudinal section} = \underline{7200} \text{ pounds per square inch.}$$

It should be noticed here, that if a spherical or cylindrical shell be subjected to internal fluid stress, the shape will remain spherical or cylindrical. In other words, such shells are self-staying. Flat surfaces, on the other hand, require to be supported or stayed, as the thin plates are unable to withstand transverse forces without bulging.

Boiler plates.—Plates for the construction of the cylindrical shells of boilers are bent into shape by being passed between rolls. The best kind of joint for the meeting edges is one which will preserve the truly cylindrical form, such as a butt joint with cover straps inside and outside.

Furnace tubes are bent into shape and the meeting edges are welded. This is done in order to avoid any extra thickness of metal such as would be produced by a riveted joint, which

would be liable to be overheated under working conditions. Furnace tubes are subjected to external fluid pressure tending to produce collapse, which is liable to occur even in a perfectly cylindrical tube should the external pressure be high enough. This is due to want of perfect uniformity in the elastic properties of the metal of the tube, which would result in slight deformation from the truly cylindrical form. This having occurred, further deformation is rapidly produced and the tube collapses. Another cause of collapse is overheating of the top of the tube, should the water level in the boiler fall too low; the part heated is weaker than the rest of the tube, and is bulged inwards by the external pressure. Collapse sometimes occurs through part of the furnace tube becoming thinner in the course of time, generally due to scale formation on the water side of the tube, which leads to the plate being worn away gradually by the action of the flame.

Riveted joints.—Boiler plates are permanently connected by **riveted joints**. In the simplest form of joint, the edges of the plates **overlap**, and the rivets are closed up in a **single** row of holes. Such a joint is called a **single riveted lap joint**; if there are **two** rows of rivets—a **double riveted lap joint**. In **butt joints** the plates are brought together, edge to edge, and **cover plates** running along the seam are placed either on one or both sides.

Rivet holes are either **punched** or **drilled**. Punching injures the material of the plate round the hole, and this must be removed by **rymering out** the holes, which, in this case, are punched smaller in diameter than the rivet hole is to be, or else the plate must be **annealed** after punching. Punching must be done with the plates separate, and for this reason the holes will not come exactly opposite one another when the plates are brought together unless a special machine is used for spacing them. The holes produced by punching are slightly **conical**, and the plates are so punched that when they are put together the smaller ends of the holes are on the inside. This produces a sounder job after the rivets are closed.

Drilling does not injure the plate, and is usually done with the plates in position, so that the holes are bound to come fairly opposite one another. The slight burr raised round the edges of the holes by drilling must be removed by separating the plates after drilling and slightly countersinking the holes.

The rivets are heated before being put into the holes, and the head is then formed by hand hammers and finished by a snap, or else machines worked by hydraulic or pneumatic pressure are employed. The plates require first to be drawn tight together by bolts; the rivet then contracts and binds the plates together after the head is formed, as it cools down; the rivets are thereby put under tensile stress of an indefinite amount.

Caulking and fullering.—To secure a steam-tight joint, the edges of the plates are **caulked** or **fullered.** In the best practice, the edges of the plates are planed before being put together. In caulking, a narrow tool is employed resembling a very blunt chisel having an edge about $\frac{1}{8}''$ thick. With this tool and a hammer the boilermaker caulks the edge of the plate to a form as shown in

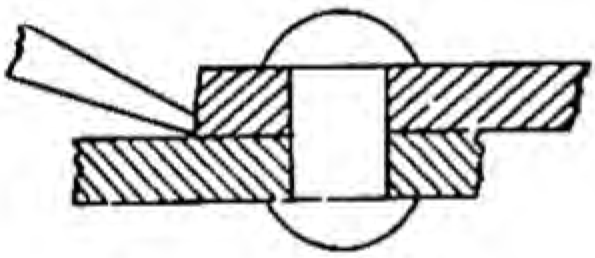


Fig. 179.—Caulking.

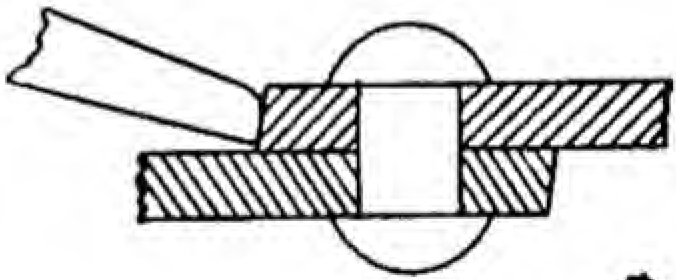


Fig. 180.—Fullering.

Fig. 179. In fullering, a broad pointed tool is used, having a thickness about the same as that of the plate (Fig. 180). In the finished work the edge of the plate is left smooth.

In the best work, fullering is employed, as the narrow groove left in caulking is liable to increase under working conditions; fullering also injures the material of the plate to a lesser degree.

Strength of riveted joints.—Considering the strength of a single riveted lap joint under pull, we see that it may fail in one of **four** ways:

(*a*) By the material between the edge of the rivet hole and the edge of plate opening out (*A*, Fig. 181). This would be due to the holes being too near the edge of the plate. In practice, it is found to be sufficient to make the distance from the edge of the hole to the edge of the plate equal to the diameter of the rivet for rivets $\frac{3}{4}''$ and greater in diameter, and slightly more for rivets less than this. Thus, for $\frac{1}{2}''$ rivets, the distance is about $\frac{3}{4}''$.

(*b*) By the material of the plate crushing at B (Fig. 181. This would be due to the rivets being too large in diameter, or to the plates being too thin. In practice, a rule such as

$$d=1{\cdot}2\sqrt{t},$$

t being the thickness of the plate in inches, is used and is found by experience to be sufficient.

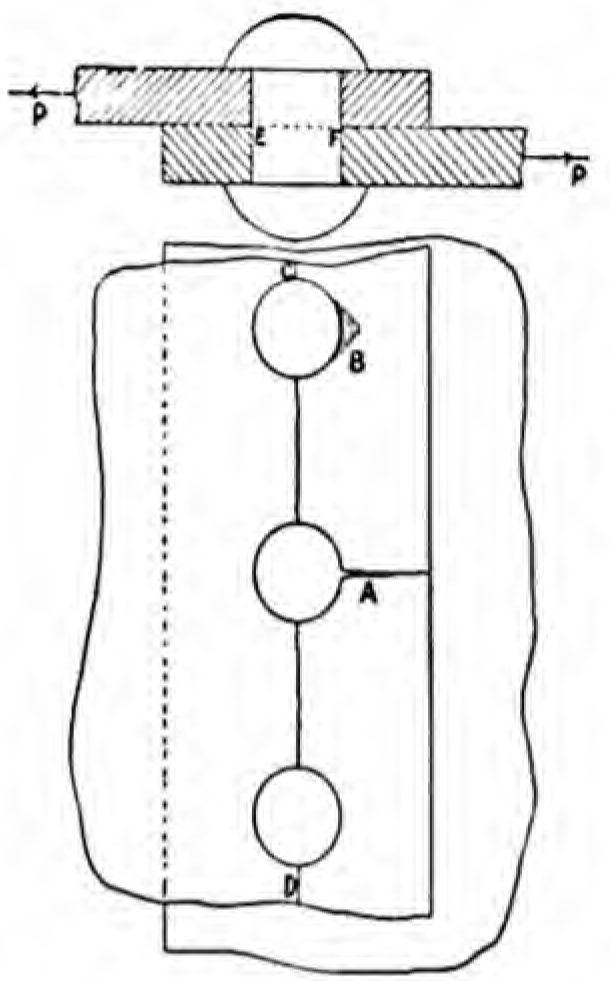


FIG. 181.—Showing the ways in which a riveted joint may fail.

(*c*) By one of the plates rupturing under tension along the line CD (Fig. 181).

(*d*) By the rivets shearing at EF (Fig. 181).

The most economical joint would be obtained by so designing it that the liabilities to rupture in these four ways are equal to one another. There being no exact mathematical information as to the strength against rupture by methods (*a*) and (*b*), it is customary to determine first the diameter of the rivet for the given plates by the rule given in (*b*), and then to decide upon the overlap of the plates as shown in (*a*). Afterwards (*c*) and (*d*) are calculated so as **to make the joint equally strong against failure by tearing along the row and by shearing of the rivets.**

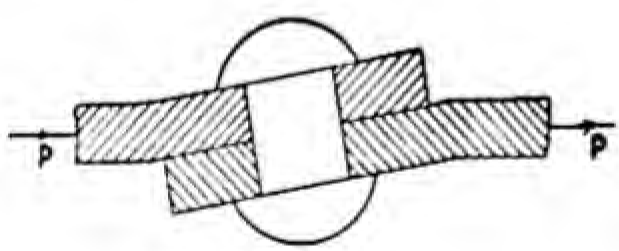


FIG. 182.

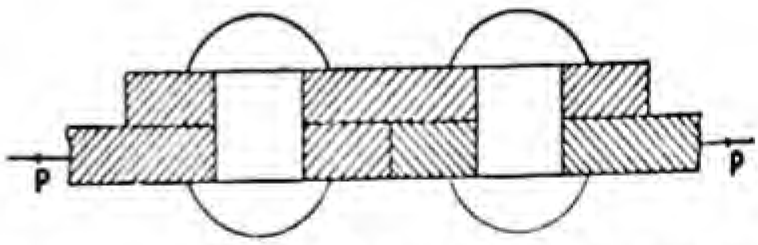


FIG. 183.—Butt joint with single cover strap.

Bending action on joints.—It must be noticed further that when the pulls P, P (Fig. 182) are applied to the joint, they produce a couple tending to make the joint assume a form resembling that shown in the illustration so that P, P will act in the same straight line. Joints are occasionally made as shown in Fig. 182 in

practice, in order to prevent this bending tendency. Butt joints are liable to the same action if there is only one cover strap (Fig. 183), but with a strap on each side (Fig. 184), this is prevented.

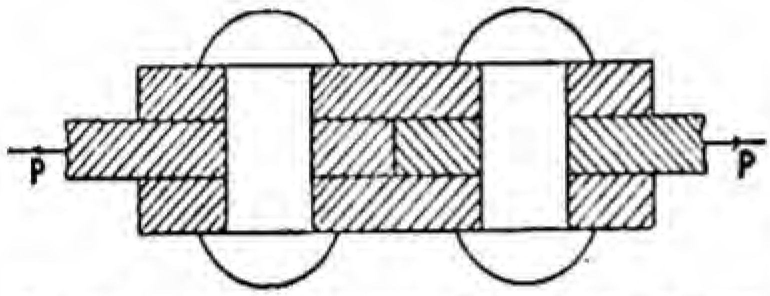


FIG. 184.—Butt joint with double cover straps.

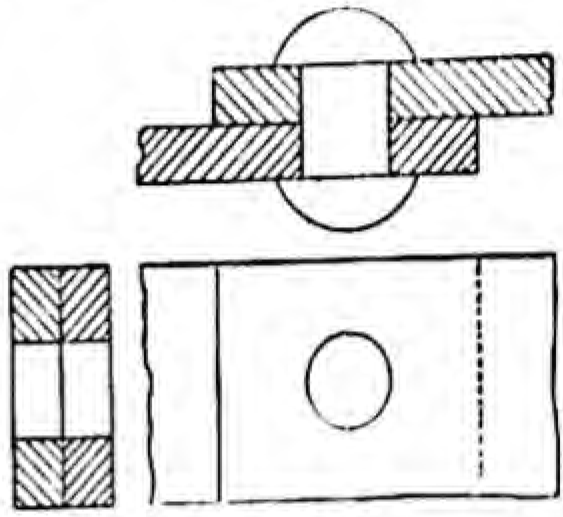


FIG. 185.

Application to single riveted lap joint.—Consider a strip of the joint equal in breadth to the **pitch** of the rivets, *i.e.* the distance from centre to centre of the rivets, measured along the row. This will be the breadth of joint supported by one rivet and so the conclusions arrived at will be true for the whole joint.

In Fig. 185 let

p = pitch of rivets, inches ;
d = diameter of rivets, inches ;
t = thickness of plate, inches ;
f_t = tensile stress permitted ;
f_s = shear stress permitted.

Area under tensile stress $=(p-d)t$.

Area under shear stress $=\frac{\pi d^2}{4}$.

For equal strength the following equation must be true :

$$f_t(p-d)t=f_s\frac{\pi d^2}{4}.$$

Taking $$d=1{\cdot}2\sqrt{t}, \text{ or } t=\frac{d^2}{1{\cdot}44},$$

then $$f_t(p-d)\frac{d^2}{1{\cdot}44}=f_s\frac{\pi d^2}{4},$$

$$f_t(p-d)=f_s\frac{\pi}{4}\times 1{\cdot}44$$

$$=0{\cdot}36 . f_s . \pi,$$

or $$(p-d)\frac{f_t}{f_s}=1{\cdot}131.$$

f_t ranges in practice from 35,000 to 67,000 pounds per square inch, and f_s from 43,000 to 53,000 pounds per square inch. The values to be taken in any given case depend on the number of rows of rivets, on the material (whether iron or steel), and on whether the holes have been punched or drilled. For iron plates and iron rivets, with drilled holes, the ratio $\frac{f_t}{f_s}$ may be taken as

$$\frac{f_t}{f_s}=0{\cdot}94,$$

which would give for the single riveted lap joint

$$(p-d)0{\cdot}94=1{\cdot}131.$$

EXAMPLE. Calculate the diameter and the pitch of the rivets for plates $\frac{1}{2}''$ thick connected by a single riveted lap joint.

$$d=1{\cdot}2\sqrt{t}$$
$$=1{\cdot}2\sqrt{\frac{1}{2}}=\frac{1{\cdot}2}{1{\cdot}41}=\frac{7''}{8} \text{ nearly.}$$
$$\left(p-\frac{7}{8}\right)0{\cdot}94=1{\cdot}131,$$
$$p=\frac{1{\cdot}131}{0{\cdot}94}+0{\cdot}875$$
$$=\underline{2\tfrac{1}{8}''} \text{ nearly.}$$

Percentage strength of joint.—It will be observed that the sectional area of the plate along the centre line of the row has been diminished after the rivet holes have been punched or drilled, and that therefore the strength of the joint is less than that of the unhurt plate. Taking a width of plate equal to p; its sectional area will be p multiplied by t in the unhurt plate and $(p-d)t$ at the centre line of the row of rivets; therefore

$$\frac{\text{strength of joint}}{\text{strength of unhurt plate}}=\frac{(p-d)t}{pt}=\frac{p-d}{p}.$$

In the above example, this will give

$$\frac{2\frac{1}{8}-\frac{7}{8}}{2\frac{1}{8}}=\frac{1{\cdot}25}{2{\cdot}125}=\underline{0{\cdot}59} \text{ nearly;}$$

or the strength of the joint is about 59 per cent. of the strength of the unhurt plate.

The percentage strength of other joints may be calculated in a

similar manner. In estimating p take the line of rivet holes along which fracture of the plate by tearing will most probably occur.

In joints such as that shown in Fig. 184, the rivets are under **double shear**, that is, they would have to shear at two places if the joint gave way by fracture of the rivets. Usually butt joints with double cover straps are double or treble riveted, having four or six rows of rivets in all. In the former case, for a length of joint equal to the pitch of the rivets, there are **four** rivet sections under shear, but as these will probably not all be effective, it is customary to reckon 3·5 rivet sections only. In joints such as this a percentage strength of 75 can be obtained.

Double-riveted lap joint.—In this joint (Fig. 125) there are two rivets per pitch, consequently,

$$\text{Area under shear stress } = 2\frac{\pi d^2}{4}.$$

$$\text{Area under tensile stress} = (p-d)t.$$

For equal strength the following equation must be satisfied:

$$f_t(p-d)t = 2f_s\frac{\pi d^2}{4}.$$

Taking, as before (p. 211), $\qquad t = \dfrac{d^2}{1 \cdot 44}$,

then,

$$f_t(p-d)\frac{d^2}{1\cdot 44} = 2f_s\frac{\pi d^2}{4},$$

$$f_t(p-d) = f_s \,.\, \pi \times 0\cdot 72,$$

or

$$(p-d)\frac{f_t}{f_s} = 2\cdot 262.$$

The percentage strength of the joint will be given by

$$\text{Percentage strength} = \left(\frac{p-d}{p}\right)100.$$

Example. The plates of a Lancashire boiler 8′ 0″ diameter are $\frac{3}{4}''$ thick. The rivets are $\frac{15}{16}''$ diameter and the circumferential joints are double riveted lap. The pitch of the rivets being $3\frac{5}{8}''$, calculate the percentage strength of the joint and the tensile stress on the plates, for a working pressure of 150 lbs. per sq. inch.

$$\begin{aligned}\text{Percentage strength of joint} &= \left(\frac{p-d}{p}\right)100\\ &= \left(\frac{3\frac{5}{8}-\frac{15}{16}}{3\frac{5}{8}}\right)100\\ &= \underline{74}.\end{aligned}$$

Stress on unhurt plate

$$=\frac{\text{steam pressure} \times \text{diam. of boiler in inches}}{4 \times \text{thickness of plate in inches}} \quad \text{(see p. 205)}$$

$$=\frac{150 \times 96}{4 \times \frac{3}{4}}$$

$$=4800 \text{ lbs. per sq. inch.}$$

$$\text{Stress on plate at joint} = \frac{4800 \times 100}{74}$$

$$=\underline{6487} \text{ lbs. per sq. in.}$$

Butt joint.—Taking as an example the longitudinal joint for a Lancashire boiler shown in Fig. 124, it will be observed that there are six rows of rivets, and five rivets per pitch of the outside row. As the rivets are under double shear, there will be ten rivet sections per pitch, and eight of these may be taken as effective. The rivets are $\frac{15}{16}''$ diameter and the pitch of the outer row is $5\frac{3}{4}''$.

$$\text{Strength of joint} = \left(\frac{p-d}{p}\right) 100$$

$$=\left(\frac{5\frac{3}{4} - \frac{15}{16}}{5\frac{3}{4}}\right) 100$$

$$=0{\cdot}836 \times 100$$

$$=\underline{83{\cdot}6} \text{ per cent.}$$

The boiler is 8 feet in diameter and has to withstand a working pressure of 150 lbs. per square inch. The plates are $\frac{3}{4}''$ thick. Consider the unhurt plate of the longitudinal seam.

Stress on unhurt plate

$$=\frac{\text{steam pressure} \times \text{diam. of boiler}}{2 \times \text{thickness of plate}} \quad \text{(p. 207)}$$

$$=\frac{150 \times 96}{2 \times \frac{3}{4}}$$

$$=9600 \text{ lbs. per sq. inch.}$$

Stress on section at rivet holes

$$=9600 \times \frac{100}{83{\cdot}6}$$

$$=\underline{11{,}480} \text{ lbs. per sq. inch.}$$

Area of one rivet section

$$=\frac{\pi d^2}{4}$$

$$=0{\cdot}693 \text{ sq. inch.}$$

Effective area under shear per pitch

$$=8\times0{\cdot}693$$

$$=5{\cdot}544 \text{ sq. inches.}$$

As the shearing force on this area has to support a length of the shell equal to the pitch, we may calculate the shearing area per inch length of the shell from

$$\text{Shearing area per inch length}=\frac{5{\cdot}544}{\text{pitch}}$$

$$=\frac{5{\cdot}544}{5{\cdot}75}$$

$$=0{\cdot}964 \text{ sq. inch.}$$

The shearing force to be carried on this area may be found from

$$\text{Shearing force}=\frac{\text{steam pressure}\times\text{diam. of boiler}}{2}$$

$$=\frac{150\times96}{2}$$

$$=7200 \text{ lbs.}$$

$$\therefore \text{ Shearing stress on rivets}=\frac{7200}{0{\cdot}964}$$

$$=\underline{7460} \text{ lbs. per sq. in.}$$

EXERCISES ON CHAPTER XII.

1. A marine boiler shell is 12′ 3″ in diameter and is constructed of plates $1\frac{1}{8}''$ thick. Calculate the stress on a longitudinal section under the ordinary working pressure of 160 lbs. per sq. inch.

2. A boiler shell is 4′ 9″ in diameter, the working pressure being 180 lbs. per square inch. Calculate the thickness of plate required if the stress allowed is $4\frac{1}{2}$ tons per square inch.

3. Why are butt joints preferred to lap joints for the longitudinal seams of a boiler? Generally the circumferential joints are of the lap type; give reasons for this.

4. Contrast the effects on the plate of punching and drilling rivet holes.

5. Explain how the riveted joints in a boiler are rendered steam tight. Which method is preferable, and why?

6. Describe and give sketches showing the ways in which a riveted joint may fail. How is provision made against each kind of fracture?

7. A boiler plate is $\frac{5}{8}''$ thick and has to be connected to a similar plate by a lap joint double riveted. Find the principal dimensions of the joint assuming that the tensile and shearing stresses are equal. Calculate the percentage strength of the joint.

8. Taking the percentage strength of the longitudinal joint to be 80, calculate the thickness of plate required for a shell 8 feet diameter, working pressure 120 lbs. per square inch, stress allowed $4\frac{1}{2}$ tons per square inch.

9. A longitudinal boiler stay has to withstand a total pull of 6 tons. Calculate its diameter for a safe stress of 9000 lbs. per square inch.

CHAPTER XIII.

FUELS.

Combustion.—Substances which are simply mixed with one another may, in general, be easily separated by some mechanical process. For example, a mixture of sugar and sand can be separated by placing the mixture in water, stirring up and then allowing to settle. The sand will sink to the bottom, and the water in which the sugar will have dissolved may be drained off. Gentle evaporation will drive off the water and the sugar may be recovered.

Substances in **chemical combination** with one another cannot be so separated, such bodies being known as **chemical compounds**. Water is the commonest example of a chemical compound, being composed of definite proportions of hydrogen and oxygen chemically united with one another. Air is a mixture consisting chiefly of oxygen and nitrogen. Many substances possess constituents which are able to unite in chemical combination with the oxygen of the atmosphere, the process being accompanied by the evolution of heat and light. This operation is called **combustion**. Substances which are suitable for the supply of heat to be used in commercial operations are called **fuels**. The principal combustible constituents of fuels are **carbon** and **hydrogen** together with compounds of these bodies called **hydrocarbons**.

Some important definitions.—An **element** in chemistry is a substance which has never been separated into two or more substances. Examples of elements are hydrogen, oxygen, carbon, copper, and iron. All substances are considered to be composed of very small particles, called **atoms**, which are regarded as indivisible. A **molecule** consists of a group of atoms, and forms

the smallest possible portion of a substance capable of independent existence. For example, a molecule of water is composed of two atoms of hydrogen and one atom of oxygen. **In all gases, the same number of molecules exist per cubic foot under similar conditions of pressure and temperature.**

By chemists, the elements are denoted by writing generally the initial letter only as a capital. Thus H stands for hydrogen, O for oxygen, C for carbon, one atom of each being denoted by the symbol. H_2 or $2H$ means 2 atoms of hydrogen, other numbers of atoms being indicated in the same way. If the weight of an atom of hydrogen be taken as 1, the atomic weights of some other elements are as follows:

Name of element.	Chemical symbol.	Atomic weight.
Hydrogen	H	1
Carbon	C	12
Nitrogen	N	14
Oxygen	O	16
Phosphorus	P	31
Sulphur	S	32

The constitution of chemical compounds is indicated by their **chemical formulae.** Thus the chemical formula of water is H_2O. This indicates not only that 2 atoms of H and 1 atom of O are present in the molecule, but that the weights are in the proportion of 2 parts of H to 16 parts of O. The molecules of hydrogen, oxygen, and other simple gases, are each made up of two atoms. Thus, the molecule of hydrogen is H_2 and of oxygen O_2. The molecule of water, H_2O, will occupy the same volume as that of a molecule of hydrogen, H_2. No element can be made to combine with any other elements in any proportion by weight other than its atomic weight or even multiples of its atomic weight.

Chemical equations afford the means of representing the combination of various elements or compounds with the formation of new compounds. Such an equation representing the formation of water would be

$$H_2+O=H_2O.$$

We may read this equation as:

(*a*) Two atoms (one molecule) of hydrogen when combined with one atom of oxygen give one molecule of water;

(*b*) Two parts by weight of hydrogen when combined with sixteen parts by weight of oxygen give 18 parts by weight of water. (Notice that as nothing can be destroyed in any chemical operation, the weights on the two sides of the equation must be equal.) The pound avoirdupois may be used as the unit of weight.

In reading (*a*), volumes may be substituted for molecules, because all gases at the same pressure and temperature have the same number of molecules per cubic foot. Thus, if one cubic foot of oxygen be used, there will be a certain number of oxygen molecules, and, to obtain double this number of hydrogen molecules at the same pressure and temperature, two cubic feet of hydrogen must be used, so that the equation may be read:

(*c*) Two cubic feet of hydrogen when combined with one cubic foot of oxygen at the same temperature and pressure give two cubic feet of gaseous water.

The engineering student should be cautioned here to write always **water**, not H_2O, when he means the liquid. H_2O may exist as ice, water, or steam.

Other chemical equations useful in the study of fuels are

$$C + O = CO.$$

We may take this equation to mean that 12 lbs. of C when combined with 16 lbs. of O give 28 lbs. of CO. CO, **carbon monoxide**, or **carbonic oxide**, is a highly poisonous gas produced by the partial combustion of carbon with a limited supply of oxygen. To burn 1 lb. of carbon until it is completely converted into carbon monoxide requires, as indicated by the above proportions, $1\frac{1}{3}$ lbs. of oxygen.

Carbon also combines with double the quantity of oxygen, the reaction being expressed by the equation

$$C + O_2 = CO_2.$$

Expressing this equation as weights, the student will find that to burn 1 lb. of C to CO_2 requires $2\frac{2}{3}$ lbs. of O. CO_2, known as **carbon dioxide**, or **carbonic acid gas**, is the gas produced by the complete combustion of carbon, which may be secured by a plentiful supply

of oxygen. It will not support combustion, and produces death in animals by suffocation.

CO is combustible, producing CO_2 as represented by the equation

$$CO+O=CO_2.$$

The combustion of sulphur is represented by the equation

$$S+O_2=SO_2,$$

the compound SO_2 is known as **sulphur dioxide.**

The student should express in weights the reactions represented in the two above equations for himself.

CH_4, **methane** or **marsh gas**, and C_2H_4, **ethylene** or **olefiant gas**, are the most important hydrocarbons with which we have to deal.

Rates of combustion.—Hydrogen, when mixed with its proper proportion of oxygen, may be exploded with violence. Hydrocarbon gases are also explosive when mixed with oxygen. Carbon burns fairly slowly unless it is finely powdered and mixed as a dust with oxygen, in which case an explosion may be produced. Carbon monoxide, CO, is also explosive when mixed with oxygen.

Amospheric air.—The atmosphere forms the principal supply of oxygen necessary to support combustion. The proportions of oxygen and nitrogen in the atmosphere are roughly 4 parts of nitrogen to 1 part of oxygen by volume, or 56 parts of nitrogen to 16 parts of oxygen by weight. More accurately, the proportions are

79 of N to 21 of O by volume ;
77 of N to 23 of O by weight.

Taking the rough numbers, it will be seen that if we require one cubic foot of oxygen in any combustion we must supply 5 cubic feet of air. If 16 lbs. of oxygen are required, we must supply $(16+56)=72$ lbs. of air, or $\frac{72}{16}=4\frac{1}{2}$ lbs. of air must be supplied if 1 lb. oxygen is required, or 4·35 lbs. using the more exact data.

Nitrogen does not aid or hinder combustion in any way other than it gets heated in the process, and so carries off a considerable portion of the heat developed, thereby preventing a higher temperature from being attained.

Air required for combustion.—The quantity of air required for the combustion of a substance may be estimated from its

chemical composition. The result may be called the **theoretical quantity** of air. Actually, from 50 to 100 per cent. more must be given to ensure complete combustion.

EXAMPLE. Calculate what weight of air must be supplied for the complete combustion of 1 lb. of carbon.

$$C + O_2 = CO_2.$$

12 lbs. carbon require 32 lbs. oxygen.

1 lb. carbon requires $2\frac{2}{3}$ lbs. oxygen.

For 1 lb. oxygen, $4\frac{1}{2}$ lbs. of air are required.

$$\therefore \text{ Air required} = 2\tfrac{2}{3} \times 4\tfrac{1}{2}$$
$$= 12 \text{ lbs.}$$

In practice, from 18 to 24 lbs. of air would be required.

Heating values.—The heat available from the combustion of 1 lb. of a fuel is called its **heating value**. The heating value of hydrogen may be taken as 62,000 B.T.U. or 34,500 lb.-deg.-Cent. units; and of carbon 14,500 B.T.U. or 8040 lb.-deg.-Cent. units when completely burned to CO_2. Carbon burned to CO gives 4450 B.T.U. or 2470 lb.-deg.-Cent. units. CO burned to CO_2 gives 10,080 B.T.U. or 5600 lb.-deg.-Cent. units. Sulphur has a heating value of 4060 B.T.U. or 2250 lb.-deg.-Cent. units.

Fuels.—Fuels are either solid, liquid or gaseous. The principal fuels in use are **coal** (solid), **petroleum** and **paraffin oil** (liquid), **lighting gas** and **power gas** (gaseous).

The value of a fuel for steam raising purposes is estimated from its heating value, percentage of volatile constituents and of ash, ease with which it can be handled, stored and fired, and its cost.

Solid fuel.—Coal consists of fossilised vegetable matter. The vegetation of past ages being buried in the earth undergoes compression and slow mineralisation; the first product is **lignite**, a coal which ranges in colour from yellow to dark brown, and is still rich in volatile constituents. Further mineralisation produces **bituminous coal**, a fuel which contains a large proportion of volatile matter. **Anthracite** is a coal produced by the elimination of the volatile constituents, and is the most perfectly mineralised coal we have. **Semi-bituminous** and **semi-anthracite** are coals intermediate in composition.

Principal constituents of coal.—These are carbon, hydrogen

and oxygen. The best qualities of anthracite contain more than 90 per cent. of carbon; bituminous coal contains from 50 to 60 per cent. of carbon and up to 30 per cent. of volatile matter. Semi-bituminous coals have proportions of carbon and volatile matter between these. The percentage of oxygen ranges from about 1·5 in the best anthracite to 30 or 40 in the poorest lignite. Ash varies from 3 per cent. in the highest classes of coal to 25 or 30 per cent. in the poorest.

The volatile matter in coal is composed chiefly of hydrocarbons. These make the coal easy to burn, and facilitate the production of flame. To burn hydrocarbons completely a high temperature and a plentiful supply of air are simultaneously required. Failure to secure these conditions with bituminous coal will cause black smoke to be formed. Much ash causes the furnaces to become dirty, and clinker may be formed by the fusing of the ash, with the consequent choking of the air spaces in the furnace. Sulphur in coal is deleterious to the furnace plates.

Anthracite is usually jet black in colour and is very clean. Sudden application of heat causes it to break up into small pieces. It is best worked with light fires and forced draught, and the furnace should be of ample capacity, as the heat is very intense. There is no smoke produced during the combustion.

Semi-anthracite and **semi-bituminous** coals are much used for steam raising purposes. The best Welsh coal contains just sufficient volatile matter to secure easy combustion without the production of black smoke. Other qualities require more care in the stoking to secure this result.

Bituminous coals are useful for the manufacture of gas, for which purpose the large quantity of volatile matter present makes them valuable.

Lignite may contain from 60 to 70 per cent. of carbon. The poorer qualities, containing as little as 30 per cent. of carbon, are not suitable for use in boiler furnaces.

Other solid fuels are **coke**, produced from coal by destructive distillation so as to drive off the volatile constituents—coke is the residue, consisting of carbon and ash; **wood**; **charcoal**, produced from wood by driving off moisture and volatile matter, leaving practically pure carbon; **peat**, which is the remains of comparatively recent vegetation found in boggy soils.

A few heating values of solid fuels are here given.

HEATING VALUES OF SOLID FUELS.

Fuel.	Heating Value, B.T.U. per lb. of fuel.
Best Welsh coal,	16,000
Average Newcastle coal,	14,900
Average Derbyshire coal,	13,900
Average Lancashire coal,	13,900
Average Scotch coal,	14,200
Lignite,	11,500 to 14,000
Fairly dry peat,	10,000
Coke,	13,000
Wood,	about ¼ that of coal
Indian Coal	10000 about

Liquid fuel.—Mineral oils suitable for fuels are obtained as (*a*) **crude petroleum**, (*b*) **paraffin oil.** These oils consist of mixtures of various hydrocarbons.

Crude petroleum is discharged from natural reservoirs in the earth's crust through wells which are drilled in likely places until oil is struck. The bulk of the supply comes from the United States and Russia. Crude petroleum is refined by a process of distillation, a number of different substances coming off, and leaving finally a thick residue. Gasoline, burning oils, oils suitable for gas making, lubricating oils, paraffin wax, etc. are thus produced. Crude petroleum and the residue from the distillation process may be used instead of coal in boiler furnaces for steam raising purposes. The light gasoline oils and the heavier burning oils are suitable for use in the cylinders of internal combustion engines.

Oils of low specific gravity (0·6 to 0·75) such as **gasoline, petroleum spirit, benzoline** require careful handling, as at ordinary atmospheric temperatures they continually give off inflammable vapour. A light brought near any of them, but not in actual contact, will cause the oil to take fire. Special precautions must therefore be taken in their storage, and in no circumstances should a naked light be introduced to the store-room. Oils of this class are largely

used for motor-cars and for this purpose are very suitable, being very easily brought into the state of vapour. The vapour, when mixed with a proper proportion of air, forms an explosive mixture to be used in the engine cylinder.

The heavier burning oils, specific gravity 0·78 to 0·828, are safer to handle. Oils such as Royal Daylight (American) and Russolene (Russian) do not give off inflammable vapour at ordinary atmospheric temperature. These oils may be easily vaporised by first finely dividing them by spraying, and then heating the spray. The resulting vapour, when mixed with air, gives the required explosive mixture.

Paraffin oil is manufactured by distilling bituminous shales and boghead coal. One of the best known seams of such shale is at Broxburn, near Edinburgh.

Flash point.—The temperature at which an oil begins to give off inflammable vapour is called its **flash point**. In this country, the legal test is performed in the Abel apparatus. The apparatus consists of a cup to contain a certain quantity of the oil under test when filled up to a standard mark. The cup is surrounded by a bath of water which is adjusted at 130° F. before the cup is placed in position. The cup is closed by a cap, furnished with a small slide which may be drawn open by hand, the same action lowering a tiny gas jet into the interior of the cup. A pendulum 24″ long is supplied with the apparatus. Three swings of the pendulum give the proper time for opening the slide and one swing for closing it. Thermometers are inserted in the water bath and into the space in the cup just above the oil surface. As the temperature of the oil rises and attains 66° F., the slide is drawn as described, and this is repeated at every subsequent degree until a flame is observed to spread inside the cup when the slide is drawn. The temperature at which this occurs is called the flash point of the oil by Abel's close test. (*Close* because a closed cup is used.) No oil may be sold for illuminating purposes in this country which has a flash point by this test lower than 73° F. This is equivalent to a flash point of about 100° F. when an open cup is used.

Properties of some well-known oils.—Royal Daylight oil has a flash point of 81° F., Russolene, 88° F., and White Rose (American), 105° F.

The composition of refined petroleum may be taken as averaging 86 per cent. of carbon and 14 per cent. of hydrogen. Russolene oil has a specific gravity of 0·825 and its heating value is about 20,300 B.T.U. per lb. Royal Daylight oil has a specific gravity of 0·797 and a heating value of 20,100 B.T.U. per lb.

Petrol of specific gravity 0·678 has a heating value of 19,800 B.T.U. per lb. Crude petroleum, American, has a specific gravity 0·886, Russian, 0·938. The heating value varies from 19,000 to 19,500 B.T.U. per lb.

Petroleum refuse has a specific gravity from 0·906 to 0·928. Its heating value averages 19,000 B.T.U. per lb.

Gaseous fuel.—The fuels of this class in common use are (*a*) ordinary lighting gas, (*b*) producer gases manufactured specially for power purposes, (*c*) gases produced as by-products from other processes, (*d*) natural gas.

Ordinary lighting gas is produced by heating bituminous coal in closed retorts, when the volatile constituents are driven off, and, after purification, are available for lighting and heating. The contents of the retorts at the completion of the process consist of coke. Lighting gas is often enriched by the addition of oil vapour. The constituents of lighting gas vary considerably in different localities. Ordinary proportions by volume are:

Constituent.	Percentage volume.	Constituent.	Percentage volume.
H_2	51·81	CO	8·95
CH_4	35·25	O_2	0·08
C_2H_4	3·53	N_2	0·38

The heating value ranges from 600 to 750 B.T.U. per cubic foot. The quantity of gas produced from a ton of coal varies from 7000 to 15,000 cubic feet, depending on the kind of coal used. 10,000 cubic feet may be taken as an average yield.

Producer gases.—**Dowson gas** is produced by blowing a mixture of air and superheated steam through incandescent anthracite or coke. This fuel is used to prevent choking of the producer by tarry matter and clinker. The resulting gas is composed chiefly

of hydrogen, carbonic oxide, and nitrogen. The average proportions by volume are :

DOWSON GAS.

Constituent.	Percentage volume.	Constituent.	Percentage volume.
H_2	18·73	CO_2	6·57
CH_4	0·31	O_2	0·03
C_2H_4	0·31	N_2	48·98
CO	25·07		

The average heating value is 160 B.T.U. per cubic foot. Modern gas engines use about 75 cubic feet of Dowson gas per indicated horse-power per hour, which works out to a consumption of about 1 lb. of coal per horse-power hour.

Mond gas is produced from bituminous slack by blowing air saturated with steam at a temperature of 70° C. through the coal which is kept burning at a dull red heat. The combustion is effected without the production of clinker or tarry matter, and the whole process is designed to avoid loss of heat. The average proportions by volume are :

MOND GAS.

Constituent.	Percentage volume.	Constituent.	Percentage volume.
H_2	28	CO_2	15
CH_4	2	O_2	—
C_2H_4	traces	N_2	43
CO	12		

The average heating value is 160 B.T.U. per cubic foot. About 70 cubic feet per indicated horse-power per hour are used in gas engines. About 150,000 cubic feet of gas are obtained from 1 ton of bituminous slack.

Suction gas plants have recently come into considerable use for supplying gas to small engines of about 10 to 50 H.P. Anthracite or coke is burned slowly in a small producer, using about 1·1 lb. per hour per brake horse-power. The air required

to maintain the combustion is drawn through the producer by the action of the engine piston during the suction stroke. The air is heated and charged with water vapour before entering the producer, where chemical action results in a gas being given off containing roughly 29 lbs. of CO, $1\frac{1}{2}$ lbs. H, 57 lbs. N, and 12 lbs. of CO_2 per 100 lbs. The gas is then cooled and is drawn into the cylinder together with the further proportion of air required to form an explosive mixture.

Oil gas is manufactured by treating heavy mineral oils. **Water gas** is produced by blowing superheated steam through incandescent anthracite or coke.

By-product gases.—**Blast-furnace gas** is given off from blast-furnaces during the smelting of iron; this gas may be used in boiler furnaces or in the cylinders of gas engines. The gas given off from coke ovens during the manufacture of coke from bituminous coal may also be used for heating purposes.

Natural gas occurs largely in the United States. Wells are bored until gas is reached, when it is discharged at a high pressure and is available for direct use for lighting and power purposes. The principal constituent is marsh gas. Pittsburg natural gas has an average heating value of 1000 B.T.U. per cubic foot.

Calculation of heating value.—The heating value of a fuel may be calculated from its composition as found by a chemical analysis. To carry out such an analysis requires the services of a highly skilled chemist, and the results obtained from the subsequent calculation are doubtful. The method is to estimate the heat available from each constituent, using the heating value of the element in the calculation. If oxygen and hydrogen are both present, it is assumed that some of the hydrogen is combined with the oxygen as water, and, as no heat will be available from this part of the hydrogen, which is already completely in combination, it is deducted from the total hydrogen present, thus:

Let O = quantity of oxygen present,
H = ,, hydrogen ,,

Then, since 8 by weight of oxygen are required to form water with 1 by weight of hydrogen,

$$\left(H-\frac{O}{8}\right)=\text{hydrogen available for heating.}$$

EXAMPLE i. A sample of petroleum contains 86 per cent. of carbon and 14 per cent. of hydrogen by weight. Calculate its heating value.

In 1 lb. of the oil there will be $\frac{86}{100}$ lb. of carbon and $\frac{14}{100}$ lb. of hydrogen. The heating values of these elements are respectively 14,500 and 62,000 B.T.U. per lb.

Heat available from the carbon in 1 lb. of the fuel

$$= \frac{86}{100} \times 14{,}500 = 12{,}470 \text{ B.T.U.}$$

Heat available from the hydrogen in 1 lb. of the fuel

$$= \frac{14}{100} \times 62{,}000 = 8{,}680 \text{ B.T.U.}$$

Total heating value of 1 lb. of the fuel $= \underline{21{,}150 \text{ B.T.U.}}$

It will be noticed in this calculation that it is assumed that the water vapour resulting from the combustion of the hydrogen has been condensed and cooled to ordinary atmospheric temperature, giving up latent and sensible heat. In very few practical operations will this occur, and the heating value as found above consequently represents a greater quantity of heat than will be available usually in practice. Since 1 lb. H gives 9 lbs. water, the fuel in the above example will produce on combustion $\frac{14}{100} \times 9 = 1{\cdot}26$ lbs. water. Deducting say 1100 B.T.U. of latent and sensible heat for each lb. of water gives a total deduction of $1100 \times 1{\cdot}26 = 1385$ B.T.U.

The heat available in the fuel will therefore be

$$21{,}150 - 1385 = \underline{19{,}770} \text{ B.T.U. per lb. fuel.}$$

EXAMPLE ii. A sample of Welsh coal has the following percentage composition:

C - - - -	88·26
H - - - -	4·66
O - - - -	0·6
S - - - -	1·77
Ash, etc. - - -	4·71
	100·00

Calculate its heating value.

$$\left.\begin{array}{r}\text{Available hydrogen}\\ \text{in 1 lb. coal}\end{array}\right\} = H - \frac{O}{8}$$

$$= \left\{\frac{4{\cdot}66}{100} - \left(\frac{6}{1000} \times \frac{1}{8}\right)\right\} \text{ lb.}$$

$$= 0{\cdot}0458 \text{ lb.}$$

Constituent.	Heating value of constituent B.T.U. per lb.	Weight of constituent available in 1 lb. of fuel. lb.	Heating value B.T.U.
C	14,500	0·8826	12,800
H_2	62,000	0·0458	2,840
S	4,000	0·0177	71

Total heating value $=15,711$.

It is customary to neglect the small quantity of sulphur in coal in estimating the heating value.

Equivalent evaporation.—The heating value of a fuel is often stated by engineers as the weight of water in lbs. at a temperature of 212° F. which could be converted into steam at the same temperature by the application of the heat contained in one pound of the fuel. This quantity is called the **equivalent evaporation of the fuel.**

Each pound of water so evaporated would take up 967 B.T.U., consequently, for a given fuel:

$$\text{Equivalent evaporation} = \frac{\text{heating value in B.T.U. per lb.}}{967}.$$

EXAMPLE. Calculate the equivalent evaporation of a fuel having a heating value of 15,461 B.T.U. per lb.

$$\text{Equivalent evaporation} = \frac{15,461}{967}$$
$$= \underline{16} \text{ nearly.}$$

To obtain the heating value of a given sample of fuel by chemical analysis requires a skilled chemist, as has been stated, and therefore it is more customary for engineers to estimate the heating value by use of a calorimeter in which the fuel is burned in an atmosphere of oxygen, the resulting heat being imparted to a measured quantity of water.

Testing for heating value of coal.—It is by no means easy to devise a calorimeter for determining the heating value of a given sample of coal which shall be simple and at the same time certain in its action. The Darling calorimeter probably is the one which

most nearly possesses these qualities. In this instrument (Fig. 186), a weighed quantity of the powdered coal is contained in a crucible C held in clips secured to the top of a short brass tube A. A plate R is also secured to the tube, and serves for the attachment of three legs L, on which the instrument rests, and also for the support of a bell glass B which is made water and gas tight at R by means of rubber rings and a brass ring secured by thumb screws. A very short cylinder H, formed of two brass plates and a ring at the edge to keep them apart, is fixed to the bottom of A. The upper plate is perforated with a number of small holes. The top hole of the bell glass B is closed by means of a rubber stopper through which passes a central brass tube connected by a flexible tube at O to an oxygen cylinder. Two brass wires W, W also pass through the stopper and are connected at P and Q to leads, so that an electric current may be passed through a platinum or iron wire which dips into the powdered coal. The whole instrument is immersed in a measured quantity of water contained in a vessel S. On passing the current the wire will become heated and ignite the coal, the action being assisted by the oxygen. Combustion of the coal is completed under a continuous supply of oxygen, the products of combustion passing downwards through A, then through the perforations in H into the surrounding water. Passing upwards, the escaping bubbles are detained long enough by the plate R to give up most of their heat to the water before finally escaping to the surface of the water. The temperature of the water is taken before and after the combustion by a thermometer T. The following directions should be followed.

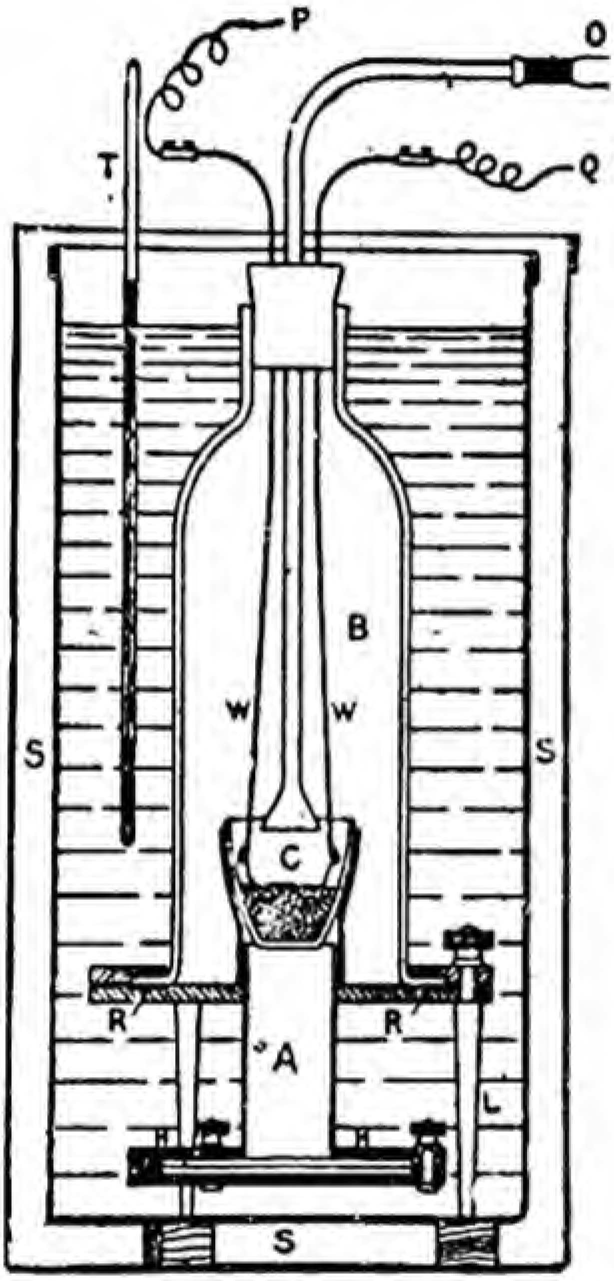


FIG. 186.—Darling calorimeter for testing the heating value of coal or other solid fuel.

EXPT. 31.—Grind up an average sample of coal in an iron mortar. Weigh out one gram in the crucible. Measure out 1400 c.c. of water into the vessel S, first bringing the temperature of the water to about 2·5° C. *below* the temperature of the room. Place the crucible in position and fasten the bell jar down. Insert the rubber stopper in the neck so that the ignition wire is embedded in the fuel. Adjust the oxygen tube so that its mouth is about ½" above the coal. Turn on a gentle stream of oxygen and immerse the apparatus in the water, taking care to adjust the pressure of the oxygen supply so that the gas may be able to overcome the head of water and so keep water out of the bell glass. Note carefully the temperature of the water and complete the battery circuit. As soon as the fuel is ignited, disconnect the battery and allow the combustion to proceed; direct the stream of oxygen so that every particle of fuel is burned; the rubber stopper permits this to be done easily. When combustion is complete, thoroughly mix the water by raising the combustion arrangement up and down until no further rise in temperature is observed. Note this temperature, then remove the combustion arrangement from the water vessel and shut off the oxygen supply.

The heating value may now be calculated.

Let Q=heating value in gram-calories per gram of coal,

W=weight of water used, grams,

W_e=water equivalent of instrument, grams,

W_f=weight of fuel burned,

t=rise in temperature of water, degrees C.

Then

$$Q=\frac{(W+W_e)t}{W_f}, \text{ gram-calories per gram of coal.}$$

$$=\frac{(W+W_e)t}{W_f}\times 1{\cdot}8, \text{ B.T.U. per lb. of coal.}$$

In the case of a platinum wire being used to start the combustion, none of it will be consumed, and the heating effect of the current used may be neglected provided care is taken to have the time during which the current is on as short as possible. If an iron wire is used, a deduction should be made for the quantity of heat from the wire which has been burned. This amounts to 1575 gram-calories per gram of wire actually burned. Sulphur may be used to start the combustion prior to placing the rubber stopper

in position; 0·05 gram of sulphur will be a sufficient quantity, and in this case a deduction of 110 gram-calories should be made from the final result for the heat evolved from this quantity of sulphur. The oxygen stream should be adjusted before placing the stopper in position, and the sulphur ignited by passing a red hot wire into the bell glass so as to touch the sulphur. Place the stopper at once into position and immerse as quickly as possible, afterwards proceeding as before.

Heating values of gaseous and liquid fuels may also be determined by use of the Darling Calorimeter in a slightly altered form to permit of the safe and complete combustion of the gas or oil. The makers are Messrs. Gallenkamp.

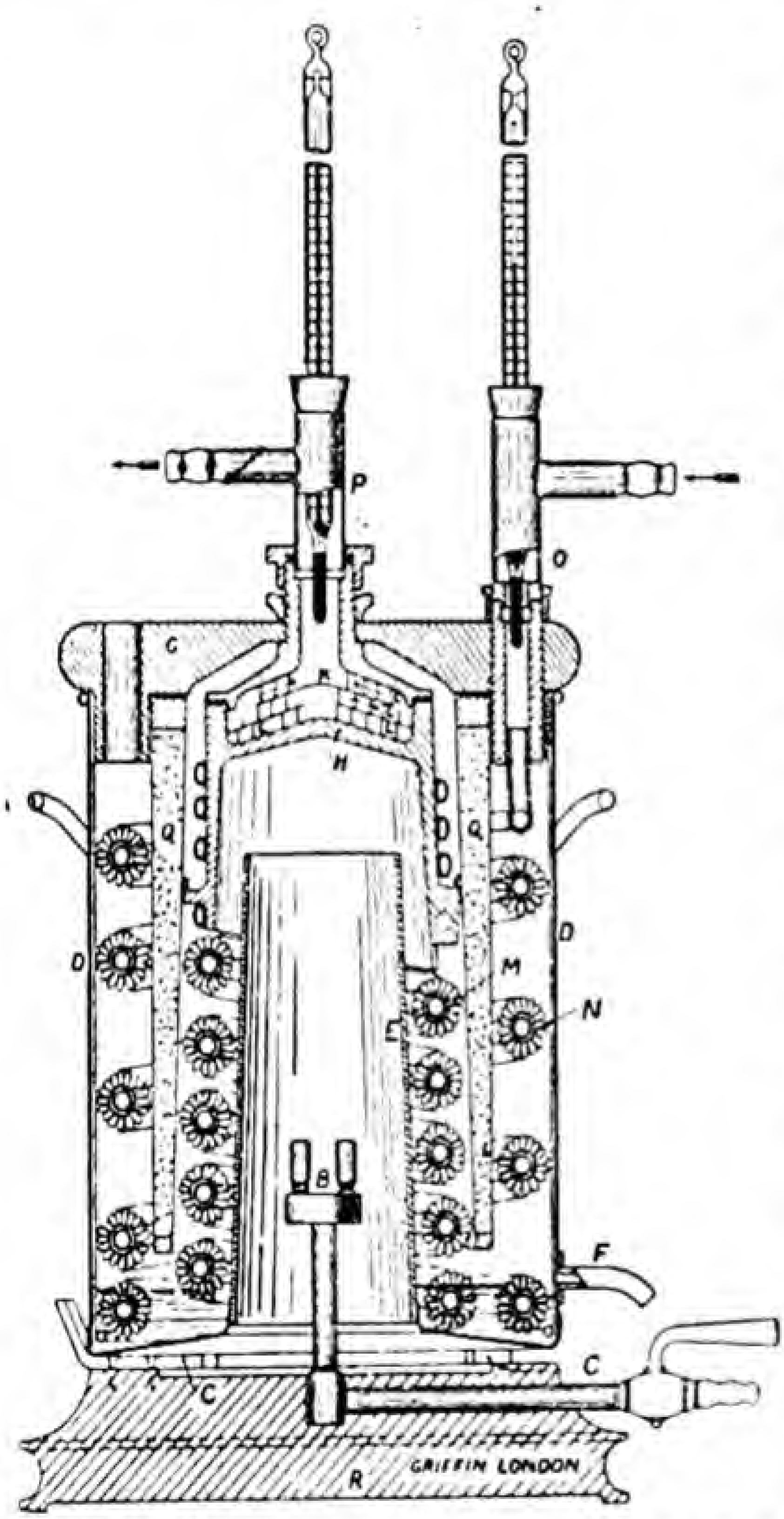


FIG. 187.—Section of Boys's calorimeter for testing the heating value of gas.

Boys's gas calorimeter.—There is no simple form of apparatus available for the beginner to employ in determining the heating value of gaseous fuel. The following apparatus is in use at the Metropolitan Gas Works, and has been devised by Prof. C. V. Boys, F.R.S. The makers are Messrs. Griffin & Sons, Ltd. The method consists in causing a steady current of circulating water to take up the heat evolved by the burning gas, and is of particular interest to engineers.

The calorimeter is shown in section in Fig. 187. The gas is

burned at two small union jets *B*, giving off products of combustion, which rise into the bell *H* and then descend outside the chimney *E*. Here the hot gases come into contact with coils containing the circulating water; the coils are made of Clarkson's motor car radiator tube, so as to be capable of rapidly abstracting the heat from the gases. The circulating water enters the outer coil at the union *O*, and, after passing through the outer coil *N* and the inner coil *M*, enters the space *K* above the bell *H*, where it circulates between two dished mixing plates, finally leaving at the union *P*. Thermometers at *O* and *P* enable the inlet and outlet temperatures of the circulating water to be measured.

Part of the products of combustion consists of steam. This steam will condense on coming into contact with the water-cooled surfaces, and the resulting water will fall to the bottom of the instrument, where a water bath is formed in which the two lower turns of the coils are immersed. The bath serves to keep the chimney *E* cool, but not cool enough to cause condensation to occur on its inner surface. An overflow *F* is provided for drawing water from the bath. The coils *M* and *N* are separated by a heat insulator *Q*, containing cork dust.

The instrument is used in conjunction with an accurate gas meter, a gas governor, and vessels for measuring the quantity of circulating water. Observations are taken of the quantity of gas, and of the quantity of circulating water with its initial and final temperature. From these data the heating value is calculated, certain corrections being applied.

Relative advantages of solid, liquid and gaseous fuels.—In contrasting the commercial value of a given fuel the following points, as well as the prime cost, should be remembered.

The ease with which coal may be procured and distributed in most districts renders this fuel by far the most useful.

With liquid fuel, dust such as is produced in storing coal is absent. Practically double the power of coal for steam raising purposes is available when liquid fuel is used, and the rate of combustion may be very easily varied to suit the given conditions. The use of oil fuel for locomotives and ship propulsion is rapidly extending.

Gaseous fuel manufactured specially on the large scale is very cheap, and distribution of the fuel through pipes to small

consumers may be effected with very little waste of energy. Engines may work economically with by-product gases which would otherwise be wasted.

EXERCISES ON CHAPTER XIII.

1. Define the terms "chemical compound," "combustion," "fuel."

2. Calculate what weight of oxygen is required to burn 3·5 lbs. of hydrogen; also the quantity required to burn completely 85·6 lbs. of carbon.

3. Describe the properties of carbon monoxide. Calculate what volume of oxygen is required to burn 100 cubic feet of carbon monoxide.

4. State the composition of the atmosphere. In Question 2, calculate approximately in each case what weight of air would be required.

5. Calculate the heat available in one ton of carbon.

6. Write a brief description of the common varieties of coal.

7. A coal contains 80 per cent. by weight of carbon, 5 of hydrogen and 10 of oxygen. Calculate its heating value.

8. In Question 7, calculate approximately the weight of air required for complete combustion of 1 lb. of the coal. What weight of air would be required in practice?

9. What is petroleum? In what forms is it used as fuel?

10. Give a brief account of any kind of gas manufactured for power purposes.

11. Define "equivalent evaporation." In a certain boiler trial, coal was used having an equivalent evaporation of 15 lbs. of water from and at 212° F. State the heating value of the coal in B.T.U. per lb.

12. In comparing the following methods of generating heat, pay attention only to cost, leaving convenience and other matters out of account.

1 lb. of average coal gives out 8500 centigrade pound heat units.

1 cubic foot of average London gas gives out 380 centigrade pound heat units.

A Board of Trade unit of electrical energy is $1\frac{1}{3}$ horse-power hours. How much heat is generated by one ton of coal? If the gas costs 3 shillings per thousand cubic feet; if the Board of Trade unit cost sixpence; what is the cost in these two cases of the amount of heat given out in burning one ton of coal. 1905.

13. A pound of oil contains 0·85 lb. of carbon and 0·15 lb. of hydrogen. What weight of oxygen is sufficient to produce CO_2 and H_2O by combustion? (Take the atomic weights of C, 12; of O, 16; of H, 1.) If 1 lb. of oxygen is contained in 4·35 lbs. of air, how many lbs. of air are needed for complete combustion? 1906.

14. A boiler furnace fire is about 12″ thick. What is known as to the way in which the combustion is going on at various places in the coal and above it and in the space just on the furnace side of the flues? Take any state you please; just before fresh coal is supplied or after, but you must say what the conditions are. 1906.

CHAPTER XIV.

EFFICIENCY.

Waste of energy.—Some of the sources of loss in the steam engine and boiler, and the means taken to produce the best results, must now be examined. The student will find it useful to consider the following calculations.

EXAMPLE i. In a certain economical steam plant, the engine gives one horse-power for an hour for each $1\frac{1}{2}$ pounds of coal fed into the boiler furnace. The heating value of the coal is 15,000 B.T.U. per lb. What percentage of the heat energy of the coal is transformed into mechanical work?

$$\text{Energy derived from 1 H.P.} = 33{,}000 \text{ ft.-lbs. per min.}$$
$$= (33{,}000 \times 60) \text{ ft.-lbs. per hr.}$$

$$\text{Energy supplied to produce this} = 15{,}000 \times 1\tfrac{1}{2}$$
$$= 22{,}500 \text{ B.T.U.}$$

Taking $J = 778$,

$$\text{Energy supplied} = 22{,}500 \times 778$$
$$= 17{,}500{,}000 \text{ ft.-lbs.}$$

$$\text{Required percentage} = \frac{33{,}000 \times 60}{17{,}500{,}000} \times 100$$
$$= \underline{11 \cdot 3}.$$

The remaining 88·7 per cent. of the heat contained by the coal has disappeared in various ways as waste.

EXAMPLE ii. In the boiler of the above steam plant, the feed water enters at a temperature of 240° F. The steam pressure is 175 lbs. per sq. in. absolute, and 11 lbs. of water are converted into steam for each lb. of coal burned. Calculate what percentage of the heat energy of the coal is taken up by the steam leaving the boiler, assuming such steam to be dry.

From the Table, p. 454, the total heat of steam at 175 lbs. per sq. in. absolute is

$$H = 1194{\cdot}9 \text{ B.T.U. (reckoned from } 32^\circ \text{ F.).}$$

As the water enters the boiler with a sensible heat given by

$$\begin{aligned} h &= 240 - 32 \\ &= 208 \text{ B.T.U.,} \end{aligned}$$

it follows that

$$\begin{aligned} \text{Heat supplied to 1 lb. of steam in the boiler} &= 1194{\cdot}9 - 208 \\ &= 986{\cdot}9 \text{ B.T.U.} \\ \text{Heat entering steam for each lb. of coal} &= 986{\cdot}9 \times 11 \\ &= 10{,}860 \text{ B.T.U.} \\ \text{Required percentage} &= \frac{10{,}860}{15{,}000} \times 100 \\ &= \underline{72{\cdot}4.} \end{aligned}$$

EXAMPLE iii. In the above steam plant, the engine produces a horse-power for an hour for every 18 lbs. weight of steam supplied. The steam is exhausted and condensed into water at a temperature of 130° F. Calculate what percentage of available energy is being converted into work, assuming the steam in the steam chest to be dry.

The steam entering the engine has a total heat of 1194·9 B.T.U., and the heat in each lb. of water leaving the condenser amounts to $(130 - 32) = 98$ B.T.U. The available energy per pound weight of steam is therefore given by

$$\begin{aligned} \text{Available energy} &= 1194{\cdot}9 - 98 \\ &= 1097 \text{ B.T.U. per lb.} \\ \text{Available energy per hour} &= 1097 \times 18 \\ &= 19{,}750 \text{ B.T.U.} \\ \text{Work produced per hour} &= 33{,}000 \times 60 \\ &= 1{,}980{,}000 \text{ ft.-lbs.} \\ &= \frac{1{,}980{,}000}{778} \\ &= 2545 \text{ B.T.U.} \\ \text{Required percentage} &= \frac{2545}{19{,}750} \times 100 \\ &= \underline{12{\cdot}9.} \end{aligned}$$

The remaining 87·1 per cent. of the heat supplied to the engine is wasted.

It may be observed from these examples that the boiler is by far the most efficient part of the plant. In the boiler the action

consists in the mere transference of heat energy from the furnace to heat energy in the steam, and such an operation is conducted with but little loss. In the engine the energy has to be converted from the form of heat into that of mechanical work, and operations of this character are always accompanied with great waste.

Causes of waste in the engine.—In the attempt to minimise so far as possible the waste of energy in the engine, engineers have been led to study the important effects of the **action of the cylinder walls** on the steam and water present in the cylinder, and also the **leakage** past the valves and piston. The student must guard against the error of supposing that the mechanism intervening between the piston and crank has anything to do with wastage other than the unavoidable waste due to frictional resistances. The principle of the conservation of energy teaches that whatever amount of mechanical work is done on the piston will appear as work done on the crank pin less that required to overcome the frictional resistances of the mechanism. The wasted power due to this cause may be calculated by taking the difference between the I.H.P. and B.H.P.

EXAMPLE. The I.H.P. of an engine is 50. The B.H.P. is 44·5. Calculate the mechanical efficiency of the engine and the energy wasted in overcoming frictional resistances.

$$\text{Mechanical efficiency} = \frac{\text{B.H.P.}}{\text{I.H.P.}} \text{ (p. 104)}$$

$$= \frac{44 \cdot 5}{50}$$

$$= 0 \cdot 89$$

$$= \underline{89} \text{ per cent.}$$

The remaining 11 per cent. of the energy supplied is wasted in overcoming frictional resistances in the engine.

$$\text{Horse-power wasted} = \text{I.H.P.} - \text{B.H.P.}$$

$$= 50 - 44 \cdot 5$$

$$= \underline{5 \cdot 5}.$$

It is found that the mechanical efficiency generally lies between 75 and 95 per cent.

Action of the cylinder walls.—Unless the steam supplied is superheated, the contents of the steam chest always consist of

steam with a comparatively small percentage of water. On admission, this mixture enters the cylinder and meets surfaces which have been cooled during the previous exhaust stroke, having been in contact with cool exhaust steam. Consequently, much of the entering steam is condensed, its heat being imparted to the walls of the cylinder. In bad cases, as much as 50 per cent. of the entering steam may be condensed before the point of cut-off is reached, giving a mixture of half-water and half-steam by weight in the cylinder at that point. During expansion, the mixture falls in pressure and also in temperature. A point will be reached beyond which, as expansion goes on, the temperature of the mixture will be lower than that of the walls. Heat will then pass from the walls into the mixture and will be expended in re-evaporating some of the water formed from the steam condensed during the admission period. On the exhaust valve opening, the contents of the cylinder will still consist largely of water, but the pressure, and consequently the temperature, of the mixed steam and water will be much lower. The action of the hot walls will therefore be very vigorous, and the water will be evaporated rapidly, thus increasing the back pressure by the generation of steam and also producing the undesirable effect of cooling the walls so that condensation will be energetic during the next admission period.

Means of limiting the action of the walls.—The action of the cylinder walls would be practically negligible if we could prevent the entry of water into the cylinder, and also its formation afterwards by condensation of some of the steam. Dry steam only should enter the cylinder, and, in order to secure this result, **separators** are often placed in the steam supply pipe close to the steam chest. This device gets rid of most of the water carried along the steam pipe. A more efficient way is to employ **superheated steam**. Such steam contains no moisture, and may be cooled considerably before the saturation temperature is reached and condensation begins. If the temperature of the superheated steam is high enough, admission and expansion may be performed in the cylinder unaccompanied by any condensation.

Should any water form in the cylinder, means must be provided for its prompt **drainage** immediately exhaust begins. It is obvious that any water present is better got rid of as water which will

carry away sensible heat only than by being boiled off into steam at the expense of the heat contained by the cylinder walls.

The **size of the engine** is an important factor. In similar engines, the volumes of the cylinders are proportional to the cubes of the diameters; and the areas of the walls are proportional to the squares of the diameters. It follows, therefore, that the action of the walls will be less important in large engines than in small ones, there being less cooling surface in the former per cubic foot of steam in the cylinder.

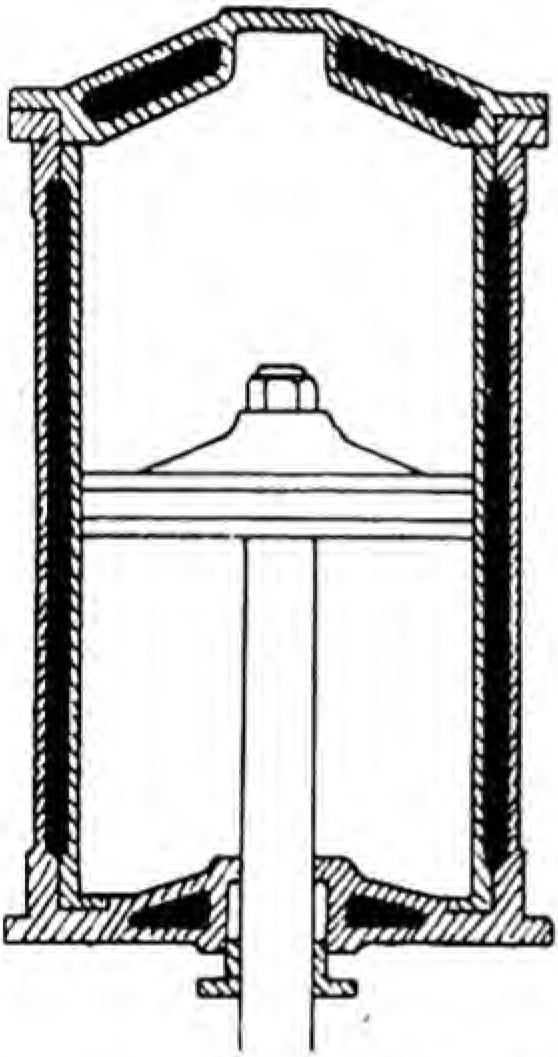


FIG. 188.—Section of a cylinder, steam-jacketed at the parts shown black.

Higher speed of revolution diminishes the action of the walls, the reason being the shorter time given for the action to take place.

Steam jackets are often employed, the method consisting of surrounding the whole of the cylinder with steam at boiler pressure (Fig. 188). The idea originally was to keep the walls of the cylinder as hot as the steam entering the cylinder. The jacket is only partially successful in effecting this, as the action takes place at the inner skin of the cylinder wall while the jacket steam is in contact with the outer skin. The action of the jacket is beneficial, however, in reducing the range of temperature to which the walls are subjected during the stroke, thus tending to reduce initial condensation during admission and also to hasten the re-evaporation. The steam in the jacket giving up its heat to the walls will undergo condensation and means must be provided for draining the resulting water away through a steam trap. In fact, a great deal of the benefit derived from the use of a steam jacket consists in removing the region of condensation from the cylinder, where drainage is imperfect, to another place, viz. the jacket, where drainage can be very perfect. Steam at boiler pressure alone should be used in the jacket. Frequently one may see, in unjacketed cylinders, the exhaust

passage from the exhaust port led round the cylinder. The exhaust steam in this passage, being charged heavily with moisture, acts very perfectly as a wet blanket to the steam in the cylinder. In the best designs this passage is taken right away from the exhaust port, and does not pass round the cylinder.

Leakage.—Leakage of steam past the slide valve into the exhaust port probably takes place by the condensation of a film of water on the port face of the cylinder. The valve slides over the film, thus uncovering it to the exhaust, where, as its temperature is higher than that corresponding to the lower pressure, it instantly boils off and is so wasted. The waste by leakage past a given valve and piston is nearly constant under given conditions of steam pressure and speed. It would show, therefore, as a higher percentage waste when the engine is cutting off early and so developing low powers, than with a later cut-off and full power. There is room for more experimental information regarding leakage for different types of valves and pistons under varying conditions. The problem is complicated by the fact that a slide valve may be quite tight when cold and yet leak considerably when heated, the effect being due to unequal expansion, causing warping of the valve or of the cylinder face.[1]

Efficiency of a perfect heat engine.—A perfect heat engine, according to Carnot's theory, is one having the following cycle of operations. The engine is supposed to take in its supply of heat from a source of heat which is kept at a constant absolute temperature τ_1, and to get rid of heat not required to another source which is kept at a constant lower absolute temperature τ_2. Thus, any heat passing into or out of the engine will do so at constant temperature, either τ_1 or τ_2, and will thus give isothermal operations. Any other operations taking place in the engine will be conducted without heat being given to or abstracted from the steam or other working substance, *i.e.* these operations will be adiabatic. Suppose the working substance to be at temperature τ_2; the cycle may be imagined to be conducted as follows: first compress the working substance adiabatically, thus raising its temperature; stop compression when the temperature is τ_1, and

[1] See *Report to the Steam Engine Research Committee*, Prof. D. S. Capper, Proc. Inst. Mech. Eng., March, 1905.

then allow the working substance to expand doing work, keeping the temperature constantly τ_1 by allowing heat to pass in from the source of heat. This operation is stopped at any desired point, and the expansion continued adiabatically. The temperature will fall, and the expansion is continued until τ_2 is reached. The working substance is now compressed isothermally, allowing heat to escape to the cold source at constant temperature τ_2 until the initial conditions are reached, when the cycle may be started again.

It may be shown that such an engine is more efficient than any other engine having a different cycle, and that its efficiency is measured by the ratio

$$\text{Efficiency} = \frac{\tau_1 - \tau_2}{\tau_1}.$$

Practically, the cycle cannot be realised and the efficiencies of all engines fall far short of this amount. The expression, however, indicates that theoretically the greater the difference between τ_1 and τ_2 the higher will be the efficiency. Practice confirms this conclusion within limits.

Means of increasing the ratio of expansion.—To obtain a large difference between τ_1 and τ_2, high boiler pressures are used (up to 300 lbs. per square inch, the temperature being 417½° F.) and the steam is expanded many times, with a correspondingly large fall in temperature. In engines discharging their exhaust into the atmosphere, the terminal pressure may be 20 lbs. per square inch absolute, giving a terminal temperature of 228° F. If a condenser be employed, the terminal pressure and temperature may be lowered to 4 lbs. per square inch and 153° F., thus enabling larger ratios of expansion to be used.[1]

To convert a non-condensing engine into a condensing engine, a condenser and air pump must be added. The condenser may be a **surface** or a **jet** condenser. In the first type, which is shown in diagram form at *D* in Fig. 189, the exhaust steam is brought into contact with the outer surface of a large number of tubes *E*, kept cool by causing water to circulate through them, the water entering at *F* and being discharged at *G*. A pipe *H* leads to the air

[1] Recently in some motor car engines having steel cylinders 4″ diameter, steam pressures up to 1000 lbs. per square inch have been employed.

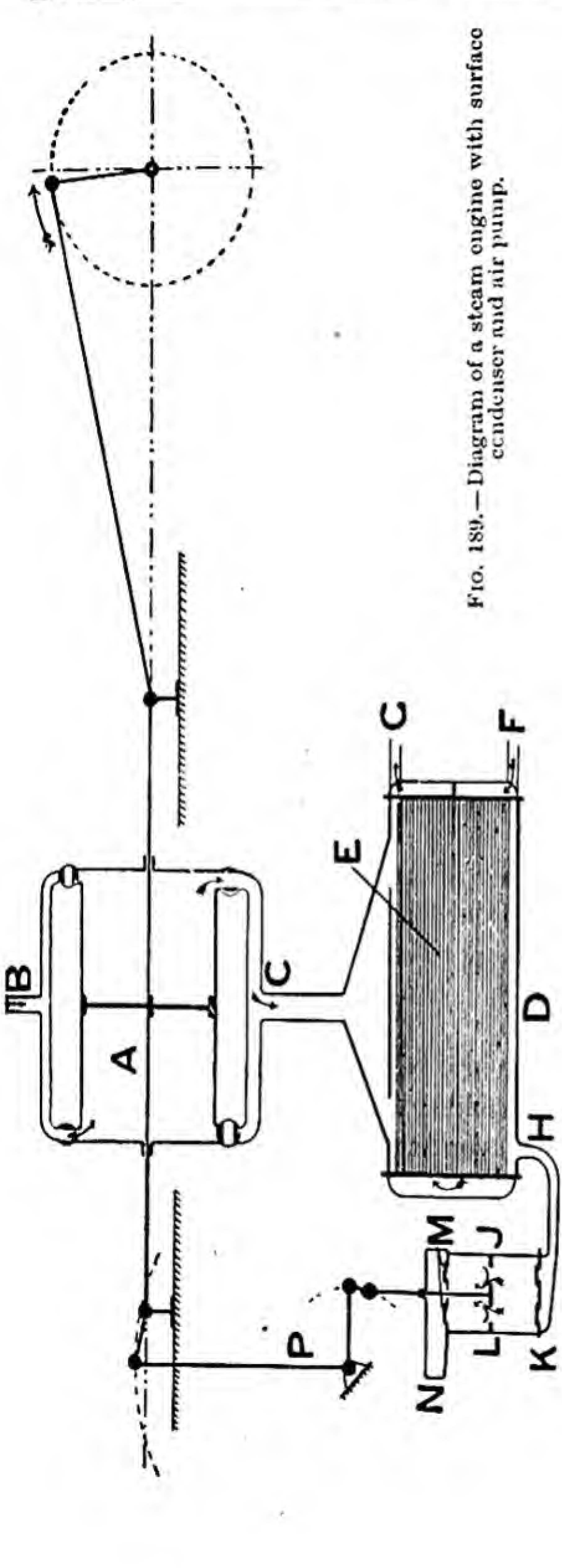


FIG. 189.—Diagram of a steam engine with surface condenser and air pump.

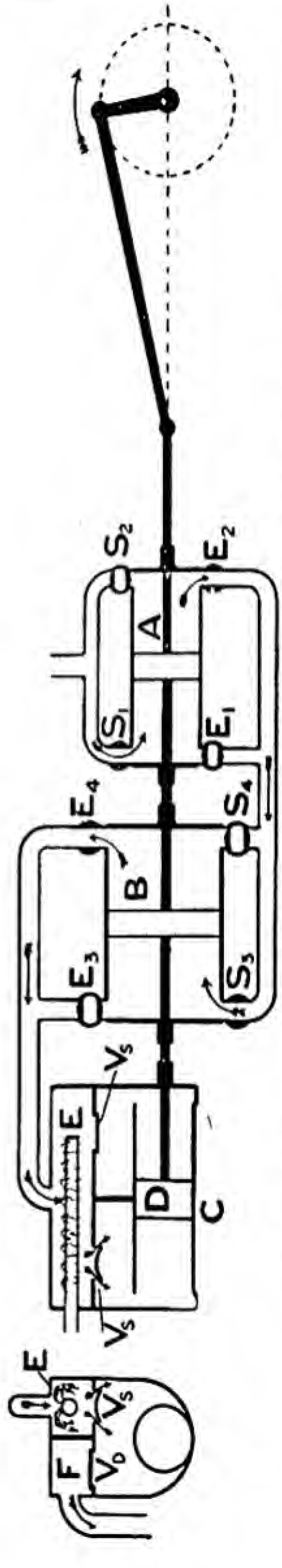


FIG. 190.—Diagram of a tandem compound condensing engine with jet condenser.

pump J, the function of which is to clear the water of condensation and air from the condenser. The air is contained in the feed water, passes from the boiler with the steam and is finally discharged into the condenser. The air pump has foot valves at K, bucket valves at L and head valves at M. The water is discharged by the air pump into the hot well N and is pumped by the feed pump back to the boiler. The air pump shown is driven from the tail end of the piston rod by means of a bent lever P.

In the jet condenser (Fig. 190, E) a spray of cold water is played on the entering exhaust steam, thus condensing it. It will be noticed that in this form of condenser, the injection water and water of condensation mingle, while in the surface condenser they are kept apart.

Ratios of expansion higher than 2 or 3 cannot be carried out successfully in a single cylinder, as there would be much too great a difference between the piston effort at the beginning and end of the stroke, with correspondingly great differences in the turning moment, and also the range of temperature through which the walls have to pass is much too large. The condensation in such a cylinder, having a ratio of expansion of say 12, would be enormous. It is the custom, therefore, to admit the steam into one cylinder and begin the expansion therein, then to exhaust from this cylinder into another where the expansion is continued. Such engines are called **compound engines.** A third and fourth cylinder may be used, giving **triple expansion** and **quadruple expansion engines** respectively. The ratio of expansion may thus be increased to 12 or 15.

A common type of horizontal compound engine is shown in outline in Fig. 190. The cylinders are arranged **tandem,** *i.e.* in the same straight line and working on the same crank. Both pistons are mounted on the same rod. The H.P. (high pressure) cylinder is shown at A, the L.P. (low pressure) cylinder at B, and the air pump at C, the bucket of the last being also mounted on the piston rod. In Fig. 190 the piston rod is moving towards the crank on the out stroke; steam is entering the H.P. cylinder through the valve S_1, and is being exhausted from the other side of the piston into the L.P. cylinder through E_2 and S_3. The other side of the L.P. cylinder is in communication with the condenser through the valve E_4. The condenser illustrated is of the jet type,

with a double acting air pump. There are no valves in the air pump bucket D; each side of the pump is furnished with suction and discharge valves V_S and V_D respectively. The suction valves allow water and air to pass from the condenser E into the air pump, and the discharge valves communicate with the hot well F.

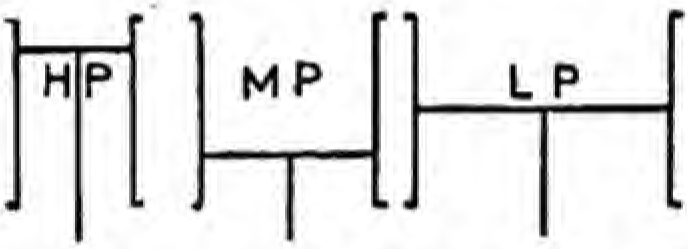


FIG. 191.—Cylinder arrangement of a three crank triple expansion engine.

Compound engines are more usually arranged with the cylinders side by side working on separate cranks. Triple expansion engines are arranged in many different ways. Figs. 191 and 192 show two such arrangements, in the first of which there are three cranks and in the second four cranks, there being two L.P. cylinders, which divide between them the steam discharged from the M.P. (intermediate pressure) cylinder.

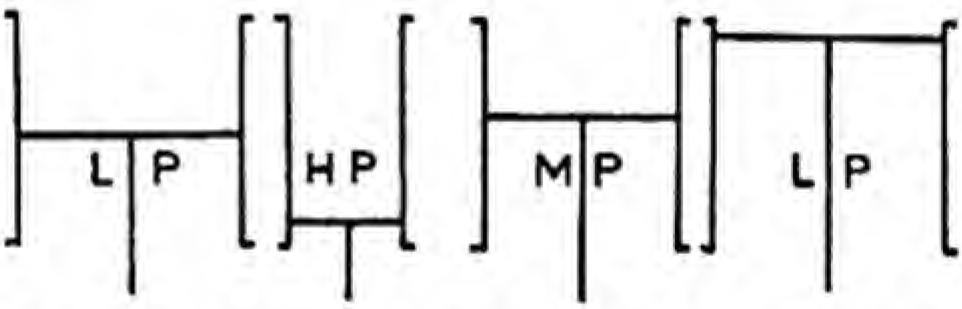


FIG. 192.—Cylinder arrangement of a four crank triple expansion engine.

Ranges of temperature and pressure.—It will be of interest to examine the ranges of pressure and temperature in the triple expansion marine engines illustrated in Fig. 202. In this engine the cylinders are 21″, 32″ and 50″ diameter respectively, with a piston stroke of 24″. The boiler pressure is 173 lbs. per square inch absolute, and the exhaust pressure is about 2 lbs. per square inch absolute. Supposing the high pressure and intermediate pressure cylinders to be cut out and the whole operation to be carried out in the low pressure cylinder, there would be an initial load of 152 tons and a terminal load of 6·15 tons on the piston. The temperature range in this cylinder would be from 369° F. to 126° F., a difference of 243° F. As the engine is arranged for

actual work, the initial working loads and ranges of temperatures are as follows:

	H.P.	M.P.	L.P.
Initial working load, tons,	12·67	17·6	18·4
Range of temperature, Fah.,	57°	58°	99°

The ratio of expansion is 8·45, neglecting clearance.

A comparison of these figures will render the advantages of multiple cylinder expansion clear.

Too large a ratio of expansion is undesirable, as little benefit will be derived from the expansion in the later stages. The last cylinder is large, and condensation and friction in it will tend to counterbalance any gain which may otherwise be obtained.

Gain from condenser.—Besides the gain which has been noted already due to the effect of the condenser in lowering the terminal pressure and temperature, the condenser gives a means of obtaining a supply of hot feed water. The water produced from the condensation of the exhaust steam in a surface condenser may be at a temperature of 120° F., and, if fed into the boiler, will require less heat to evaporate it than if cold feed water is employed. Further, this water has already been through the boiler and is therefore distilled water, *i.e.* all salts have been got rid of. Thus, in marine engines, a means is provided of feeding fresh water into the boiler instead of sea water, the salts from which would be deposited in the boiler. Feed water drawn from the condenser may contain oil which has been used for lubricating the cylinder and valves of the engines. On no account should such oil be fed into the boiler, as it would cling to the metal surfaces, and, being highly non-conductive of heat, would lead to the boiler plates being burned. To obviate this risk, an **oil separator** may be fitted and the feed water passed through it before entering the boiler.

Wasted heat in the boiler.—Some of the causes of waste of heat are due to bad stoking, giving **imperfect combustion** of the fuel and the consequent escape of combustible gases up the chimney. The admission of air to the furnace in excess of that required for complete combustion of the fuel is also a source of waste. Each pound of **excess air** is heated in passing through the furnace, where it does no good, and carries away some of the heat which would otherwise be utilised in heating the water in the boiler. The

presence of **soot**, etc. on the hot-gases' side of the heating surface, and of **scale** on the water side, **impairs the heating surface** by preventing the free passage of heat through the plates and so lowering the efficiency. **Bad circulation** of the water in the boiler prevents the free formation of steam. To obtain the best results from the heating surface, the current of hot gases should be broken up and **scrubbed** as it were against the plates, in order to extract as much heat from them as possible. Other causes of waste are in unburnt fuel dropping through the spaces in the fire bars into the ash pit, and in conduction from the surfaces of the boiler which are exposed to the air. To minimise the latter, all such surfaces should be coated as far as possible with non-conducting material.

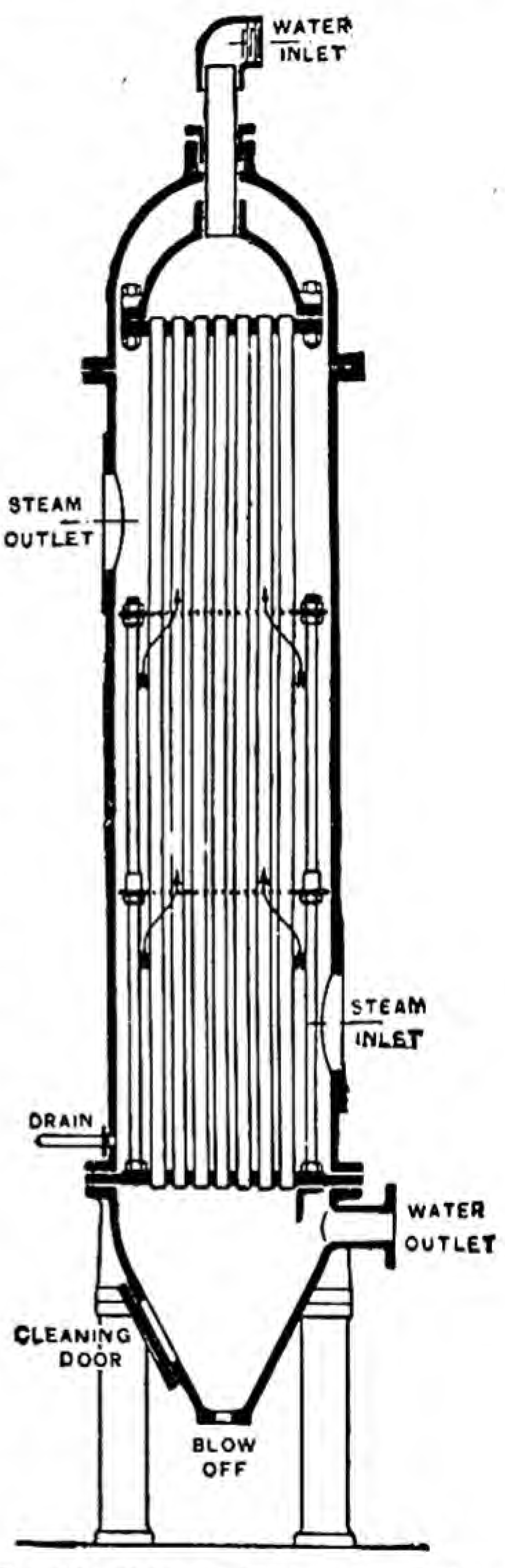


FIG. 193.—Holden & Brooke's exhaust steam feed water heater.

Feed heaters and economisers.—In surface condensing engines, the hot water from the condenser is usually employed to feed the boiler. In non-condensing engines, the exhaust steam from the engine may be used to heat the feed water by passing it through a **heater**. Fig. 193

shows such a heater in section. A number of brass tubes is arranged vertically inside an outer casing. The exhaust steam inlet from the engine enters the casing near the bottom, and the steam after circulating round the tubes is discharged near the top of the casing. The cold feed water enters at the top, descends through the tubes and is discharged hot at the bottom. The lower part of the heater is so formed as to allow any sediment in the feed water to settle in the conical receptacle whence it may be blown off periodically. Such heaters provide the means of utilising the heat which would otherwise be carried off as waste by the exhaust steam. The superior expansion of the hot tubes over the colder casing in this heater is provided for by the stuffing-box at the top, enabling the tubes and their attachments to rise and fall slightly without straining any part.

Economisers are feed-water-heaters placed in the flues between the boilers and the chimney and heated by the furnace gases. The best known type is **Green's economiser** in which there is a number of vertical tubes through which the feed water is pumped. The furnace gases pass round the exteriors of the tubes and give up some of their heat to the water. Soot and tarry matters deposited on the outside of the tubes materially affect the efficiency, and are got rid of by means of scrapers which embrace the tubes and are moved slowly up and down continuously by means of chains driven by a small engine, or other motor. Such economisers really form an extension of the heating surface of the boiler, but are more effective than would be the plan of extending the actual boiler heating surface. The rate of flow of heat through a plate depends on the difference in temperature on the two sides. The terminal heating surface in the boiler has hot water in the boiler on one side and comparatively cool gases on the other side, and is, therefore, not very effective; the difference in temperature on the two sides of the plate is not sufficient. The better plan is to extend the heating surface by fitting an economiser, in which there will be cold feed water on one side of the plates and comparatively hot gases on the other side.

Draught.—Chimney draught is often relied upon for the production of the required current of air through the furnace. The chimney operates by reason of the difference in temperature inside and outside of it. Hot air weighs less per cubic

foot than cold air. Thus, the column of gas inside the chimney weighs less than the corresponding column outside, and there will be less pressure inside at the base of the chimney than exists outside. The excess pressure of the atmosphere outside will thus cause a current to flow through the furnace towards the chimney.

In other cases chimney draught may not be sufficient to cause a high enough rate of combustion per square foot of grate area. Chimney draught may be increased somewhat by extending the height of the chimney, but there is a limit to this gain, set by the additional frictional resistances of the walls in a tall chimney, and also by the cooling of the gases. Artificial means of increasing the draught are then employed. In **forced draught**, the air is driven through the furnace by a fan or blower operated by a steam or other motor. In **induced draught**, the fan is placed at the bottom of the chimney, and operates by drawing the gases from the flues and discharging up the chimney.

Artificial draught is beneficial in securing a plentiful supply of air properly distributed in the furnace and so may produce better combustion. It is also found that the proportion of excess air can be considerably reduced by the proper application of artificial draught. As the strength of the draught is increased a higher rate of combustion may be obtained. With ordinary chimney draught, about 15 to 20 lbs. of coal can be burned per hour per square foot of grate area; this may be increased to as much as 120 lbs. with artificial draught. Should the draught be varied so much in the same boiler, it will be found that the water evaporated per pound of coal will be less with high rates of combustion, although, of course, the total water evaporated per hour will be greater. Artificial draught is much used for marine purposes, especially naval, and enables maximum power to be obtained out of the minimum dead weight of boilers.

Mechanical stoking.—With certain classes of coal being burned in large quantities in a battery of boilers, there is often difficulty in obtaining with hand firing that uniformity of conditions which is essential to efficiency. In such cases, mechanical stokers often show a distinct economy over hand firing. There are two varieties of mechanical stokers—**sprinkling** and **coking**. In the first, the coal is sprinkled over the whole of the grate. In the

second, the coal is fed to the front end, and is gradually carried to the rear of the furnace. It thus first undergoes a distillation process, the hydrocarbons being driven off and burnt and the coal converted into coke before active combustion begins.

A well-known form of coking furnace in which the worst kinds of slack or bituminous coal can be burned without the production of black smoke is shown in Fig. 194. The grate consists of a broad endless chain, consisting of many short links, and passes over two

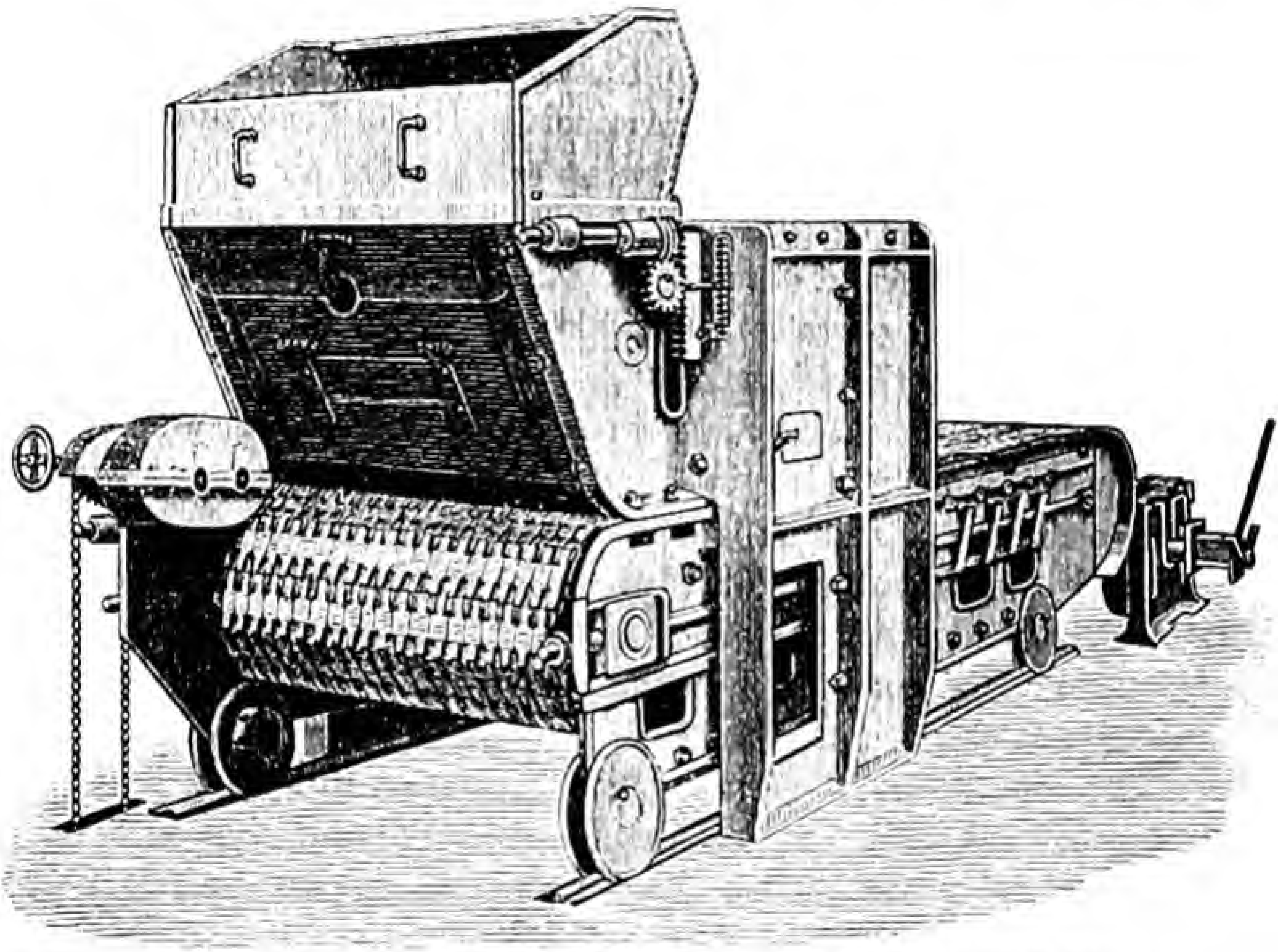


FIG. 194.—Babcock & Wilcox chain grate and coking mechanical stoker.

drums, one outside the boiler front and one inside the furnace. The front drum is revolved slowly by a worm and worm wheel, driven by a ratchet gear, having a rod actuated from a line shaft driven by a motor or small engine. The top side of the chain grate thus slowly advances inwards. Coal is supplied to a hopper and descends from it on to the grate to be carried into the furnace. The thickness of coal layer is adjusted by vertical lifting fire doors on the boiler front. Coking first occurs, then the residual coke becomes incandescent as the hotter part of the furnace is reached. By the time the fuel has reached the inside drum, combustion is

complete, and the ashes, etc., drop over the drum as the chain bends round it, and so fall into the ash pit. It will be observed that clinker and ash are cleared automatically from the furnace and thus there is no necessity for opening the fire doors for cleaning or clinkering. No cold air can enter the furnace otherwise than between the bars of the grate. Further, the bars entering the furnace are cool, and thus there is less chance of clinker forming. The whole arrangement is mounted on a carriage running on rails and can be withdrawn clear from the boiler for examination and repairs.

EXERCISES ON CHAPTER XIV.

1. In a certain steam boiler, 11 lbs. of water were evaporated per lb. of coal burned. The steam pressure was 150 lbs. per square inch absolute, and the temperature of the feed supply 120° F. The heating value of the coal was 15,200 B.T.U. per lb. Calculate what percentage of the heat supplied in the coal has been passed into the dry steam leaving the boiler.

2. Answer question 1 on the assumption that the steam passing the main stop valve is 4 per cent. wet.

3. A certain engine develops 1 I.H.P. for a consumption of steam of 15 lbs. weight per hour. Assuming the steam to be dry and at a pressure of 195 lbs. per square inch absolute in the steam chest, what quantity of heat per I.H.P. per hour is supplied to the engine?

4. Answer question 3 assuming the steam to be 12 per cent. wet on entering the steam chest.

5. In question 3, the water of condensation is discharged from the condenser at a temperature of 130° F. Calculate what quantity of heat per I.H.P. per hour is available for conversion into work. What percentage of this is actually converted into work?

6. Give a brief account of the action of the cylinder walls on the steam during (*a*) admission, (*b*) expansion, (*c*) exhaust.

7. Explain briefly the effect of the following factors on cylinder condensation, (*a*) separators, (*b*) use of super-heated steam, (*c*) drainage, (*d*) size of engine, (*e*) higher speed of revolution, (*f*) steam jackets.

8. Reciprocating engines cannot economically use a very large ratio of expansion. Explain why this is so.

9. What additional parts must be supplied to a simple engine in order to convert it into a condensing engine. Give an outline sketch and name the parts of a compound condensing engine with either jet or surface condenser.

10. Explain clearly why expansion carried out in several cylinders in series is more economical than the same degree of expansion carried out in one cylinder only. State any other practical reasons for the adoption of multiple expansion engines.

11. Enumerate the causes of wasted heat in generating steam in a boiler. Explain how these may be reduced.

12. Explain the action and give sketches of any type of feed heater or economiser.

13. Sketch and describe any type of mechanical stoker.

14. What are the objects in using (*a*) moderate forced draught, (*b*) high forced draught.

15. What do you understand by the efficiency of an engine? What would be the efficiency of a good marine engine and boiler which indicates one horse-power for every 2 lbs. of coal consumed in the furnace of the boiler per hour, supposing the coal to have a calorific power of 14,500 Fahrenheit Thermal Units per lb.? 1897.

16. How do we try to prevent condensation in a cylinder? If any of the methods serve some other good object, state it. 1905.

17. Choose any kind of boiler. Explain how by its construction, 1st, the combustion is made as complete as possible; 2nd, as much of the heat as possible is given to the water. You need not speak of careful firing.

18. Describe a mechanical stoker and how it acts. Under what circumstances is its use preferable to hand firing? 1900

CHAPTER XV.

COMPOUND AND TRIPLE EXPANSION ENGINES.

Compound engine.—The following description is of the compound engine manufactured by Messrs. Belliss & Morcom, Ltd.,

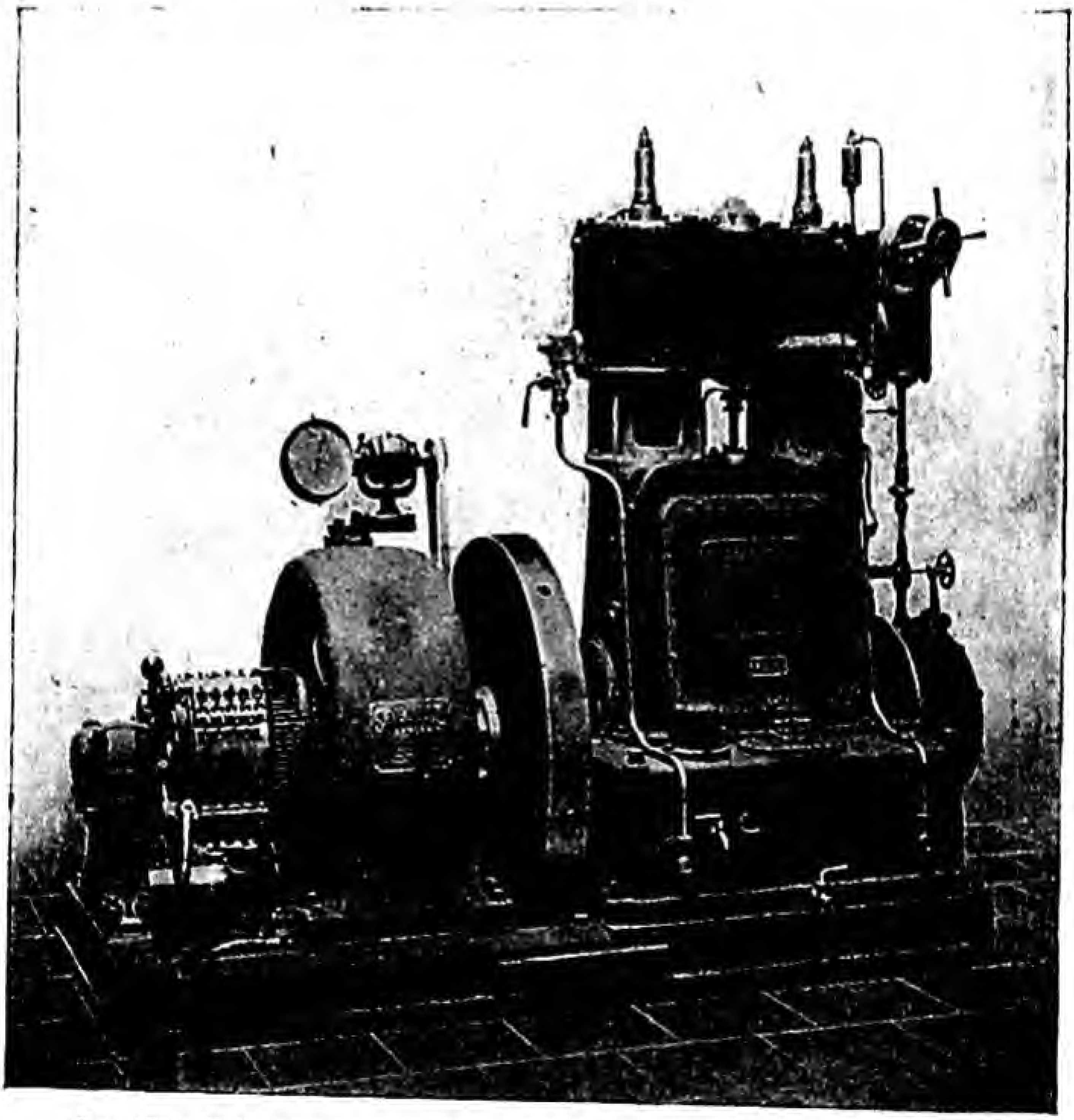


Fig. 195.—Belliss quick-revolution compound engine direct-coupled to an electric generator.

an engine which is extensively used for electric generation and other purposes. An external view of the engine is shown in

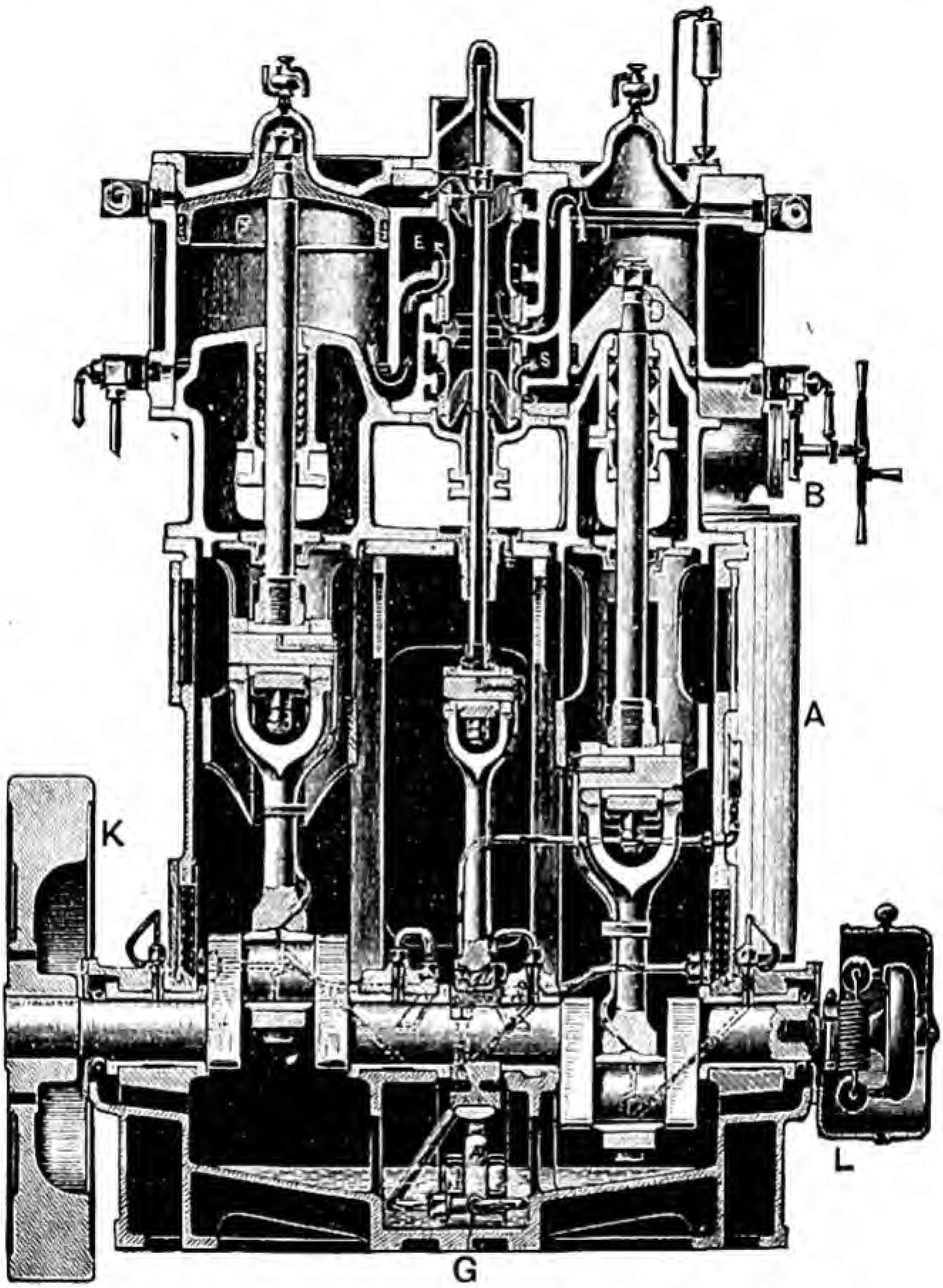


FIG. 196.—Longitudinal section of a Belliss compound engine.

Fig. 195, the engine being directly coupled to a dynamo. Sections of the engine are given in Figs. 196 and 197. The engine is of the inclosed type, having a single piston valve placed between the H.P. and L.P. cylinders. The steam distribution to both cylinders is accomplished by this valve. Forced lubrication is employed to

feed all the bearings and the engine can be run at a high speed of revolution.

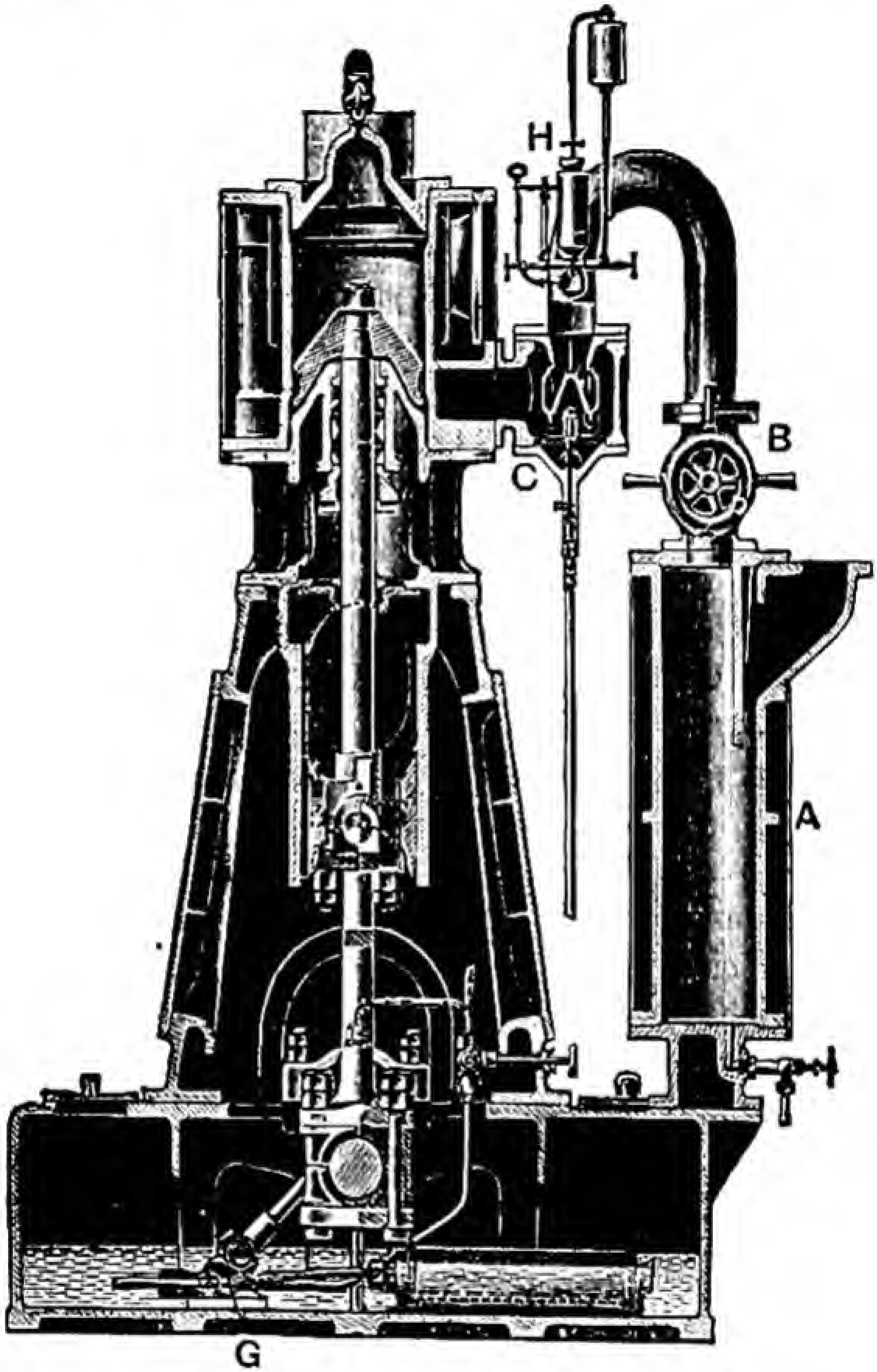


FIG. 197.—Cross section through the H.P. cylinder of a Belliss compound engine.

Referring to Fig. 197, the steam is led from the main steam pipe into a separator A, then through a stop valve B to a throttle valve C, which is controlled by the governor. The steam then enters the space S (Fig. 196). In Fig. 196 the H.P. piston D is starting on the up stroke, while the L.P. piston F is starting on

the down stroke, the cranks being arranged opposite one another. The piston valve for this position of the pistons is so situated that steam may flow from S into the H.P. cylinder below the piston, as shown by the arrow, the top side meanwhile exhausting into the interior of the piston valve, which thus serves as a receiver for the steam passing from the H.P. to the L.P. cylinder. From the receiver, the steam passes, as shown by the arrow, into the top of the L.P. cylinder, the bottom of this cylinder being in communication with the exhaust passage E which leads to the condenser. On the down stroke of the H.P. piston the distribution is as follows: steam to the top side of the H.P. piston, exhaust to the receiver from the bottom; steam from the receiver to the bottom of the L.P. piston, exhaust to the condenser from the top.

The valve is driven from an eccentric placed at the middle of the crank shaft. The same eccentric also drives a small oil pump G, which delivers oil to all of the bearings through leading pipes. The oil is at a pressure of from 10 to 20 lbs. per square inch. As the bearings are all of ample area, the pressure of the oil supply is sufficient to keep the metallic surfaces out of contact, thus the friction to be overcome is not that of metal rubbing on metal, but of metal rubbing on oil. The engines have thus a very high mechanical efficiency, and the wear is so small that cases are on record of engines running for two years and, on being opened out, found to require no taking up at any of the bearings. A sight feed lubricator for supplying oil to the piston valve and pistons is fitted at H. The crank shaft has a flywheel fitted at K and the governor at L. The shaft rotates in four main bearings formed in the soleplate, the latter being so formed as to receive all oil discharged from the bearings, whence the oil pump will deliver it again to the bearings after straining. The working parts of the engine are enclosed in a casing with movable covers. It will be noticed that the case is open just under the cylinders to give free access to the stuffing-boxes.

The governor is of the centrifugal type, consisting of two heavy balls A, A, (Fig. 198) at the ends of bent levers B, B, which are pivoted to a collar C fixed to the crank shaft. The balls are pulled inwards towards the shaft by means of springs D, D. A sliding sleeve E on the shaft is controlled by the shorter arms of the bent

levers, and will thus take up a position on the shaft depending on the position of the balls and thus on the speed of the engine. Another bent lever *F*, pivoted at *G* to a fixed axis, is operated by the movements of the sleeve *E*, and thus communicates the movements of the balls to the equilibrium throttle valve *H* through the rod *K*. The lever *F* is further loaded with a spring *L* fitted with an adjusting screw and hand wheel, thus enabling the speed of the engine to be adjusted within small limits.

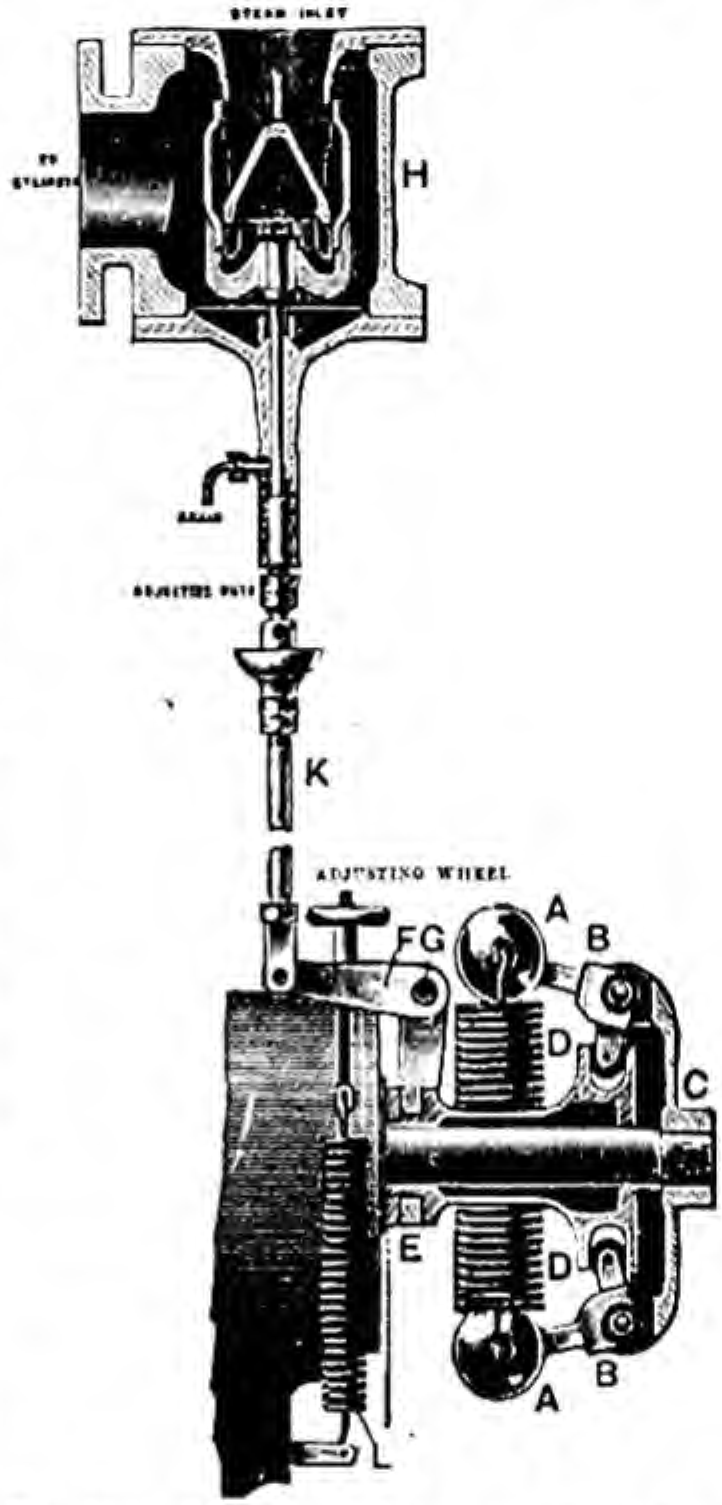


FIG. 198.—Belliss crank shaft governor and throttle valve.

The condensing arrangements are not shown in the illustrations. These are generally kept quite independent in electric generating stations, and serve to receive the exhaust steam from all the engines in the station.

Steam consumption and efficiency.—Some results of a trial on a Belliss compound engine are given. Typical indicator diagrams are shown in Figs. 199 and 200.

EXTRACT FROM LOG.

Load.	Press. at stop valve.	Press. in H.P. chest.	Vacuum.	Revols.	I.H.P.			B.H.P.	Temp. of Steam. Fah.	Consumpt. per B.H.P. hour.	Mech. Eff. $\frac{\text{B.H.P.}}{\text{I.H.P.}}$
					H.P.	L.P.	Total.				
Full	160	129	26·5″	375	167	153	320	300	490°	15·67	93·7
½	155	65	26·7″	375	87	78·7	165·7	150	480°	16·8	90·5

The engine cylinders have the following dimensions :

H.P. cylinder	-	13″ diameter
L.P. „	-	22″ „
Stroke - -	-	11″ „

It will be noticed that steam having a superheat of about 120° F. was supplied. A diagram showing the consumption of steam and the mechanical efficiency at various powers is given in Fig. 201.

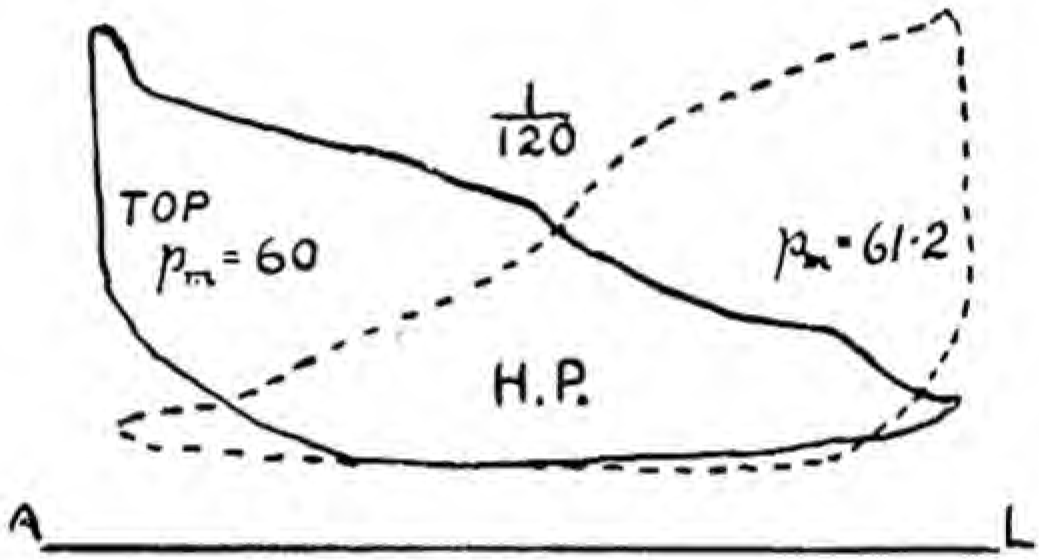


FIG. 199.—Indicator diagram from the H.P. cylinder of a Belliss compound engine.

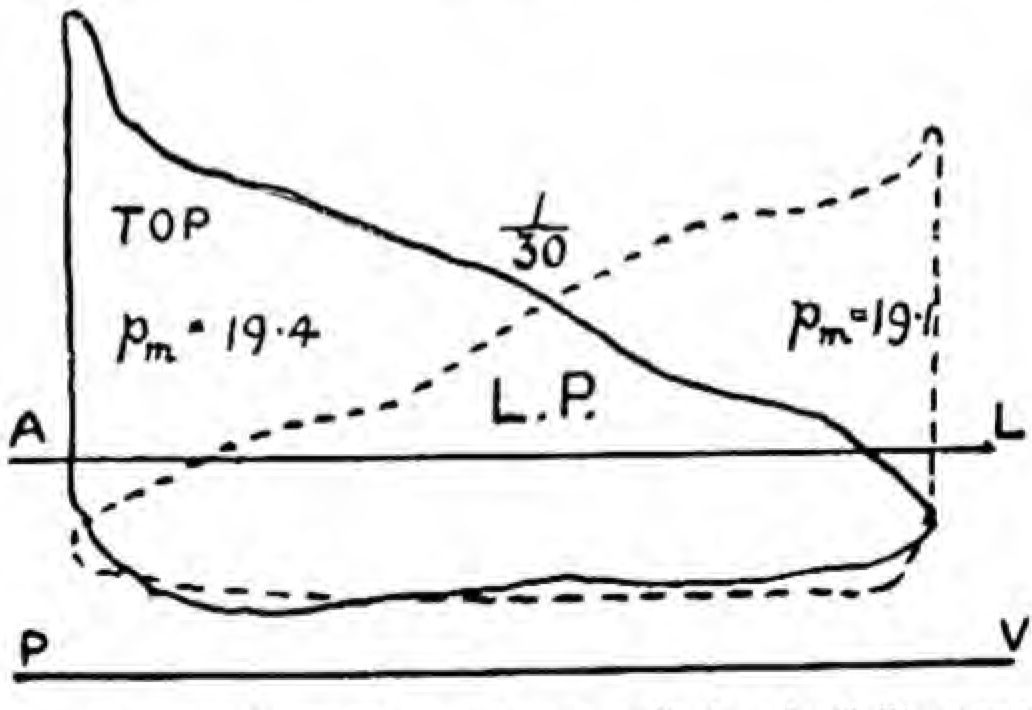


FIG. 200.—Indicator diagram from the L.P. cylinder of a Belliss compound engine.

Modes of propulsion.—Ships may be driven through the water by the action of (*a*) **paddle wheels,** (*b*) **screw propellers** placed outside the ship at the stern, or (*c*) placed inside the ship, operating by discharging jets of water through openings in the side of the ship, the method being described as **jet propulsion.**

In all these arrangements the action is the same, viz. a stream

of water is driven astern by the wheel, thus taking up momentum in the backward direction, and consequently exerting a forward pressure, which is communicated to the hull of the ship and so propels it.

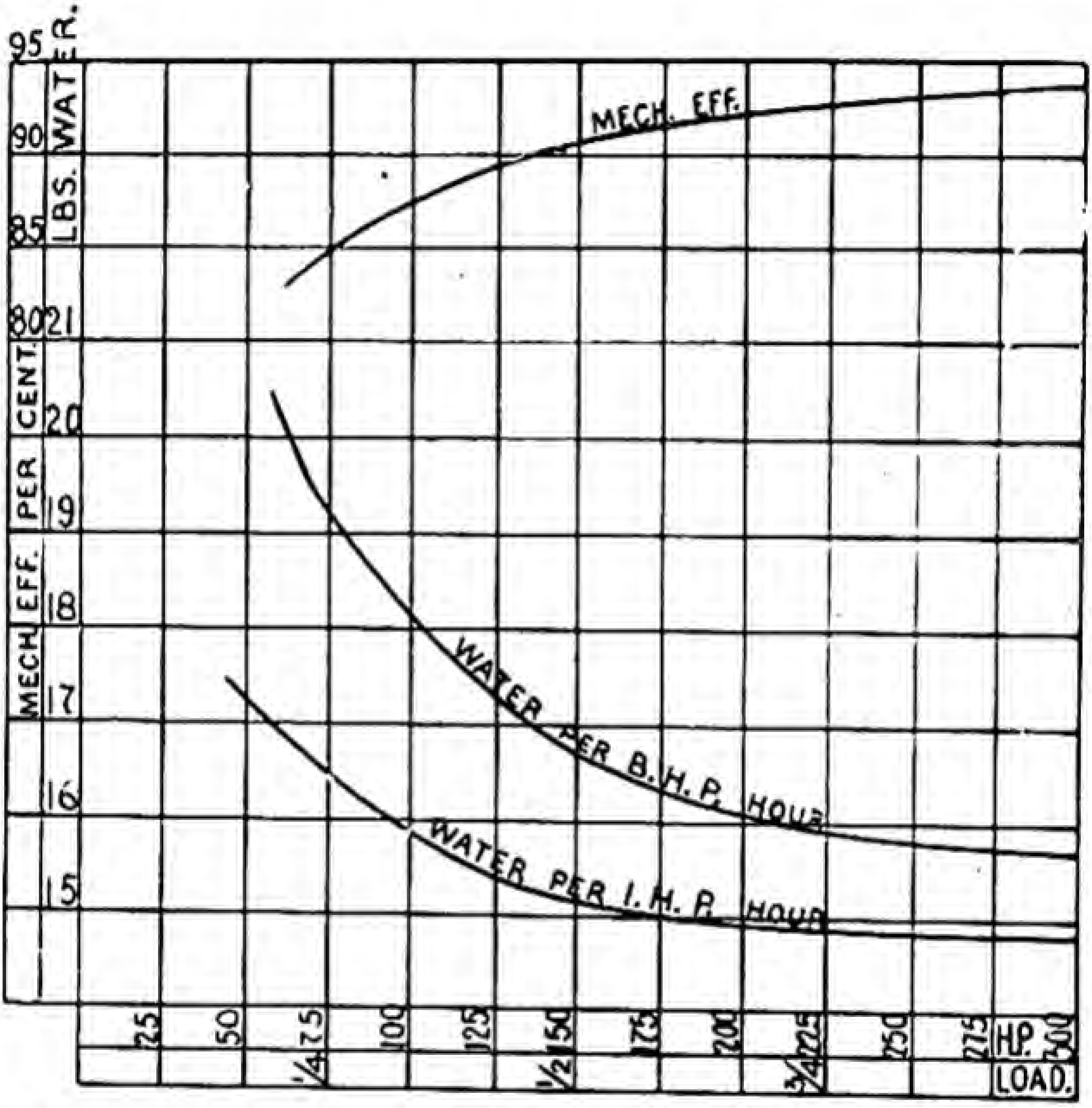


Fig. 201.—Steam consumption and mechanical efficiency of a Belliss compound engine.

Owing to the ship having to carry its own coal supply during long voyages, the skill of marine engineers has long been directed to produce the most economical results. Of the many different types of marine engines, space can only be found here for the complete description of a set of Triple Expansion Engines for screw propulsion.

TRIPLE EXPANSION MARINE ENGINES.

General arrangement.—The following description refers to a set of triple expansion marine engines with propeller, constructed for a vessel intended for transport purposes by the Thames

Engineering Company, Ltd., the drawings shown being reproduced by the courtesy of the makers. The student will already be familiar with the boilers for supplying steam to these engines from the description given in Chapter X.

The general arrangement of these engines will be understood by reference to Fig. 202, in which is given an elevation showing the cylinders in section and an end elevation. The **high pressure cylinder**, in which the steam is used first is shown at *A*. The steam is distributed to this cylinder from a valve chest *B* by means of a piston valve. The exhaust steam from this cylinder is discharged into the intermediate pressure valve chest *C*, where a double ported slide valve distributes the steam to the **intermediate cylinder** *D*. On exhausting from this cylinder, the steam is led into the low pressure valve chest *E*, where another double ported slide valve distributes it to the **low pressure cylinder** *F*. The pipes connecting the steam chests, by means of which the steam is led from cylinder to cylinder, are partly shown to the left-hand side of the cylinders in the end elevation.

On exhausting from the low pressure cylinder the steam is led through the pipe *G* into the **surface condenser** *H*. The water from the condensed steam and also any air present are drawn from the condenser by the air pump *J*, and the water is fed into the boilers by means of the feed pump *K*. These pumps are driven by levers *L* operated from the intermediate crosshead *M*. Circulating water is supplied to the condenser from the sea and is pumped overboard again.

It will be noticed that the condenser casting forms part of three columns *N*, supporting the cylinders at the back of the engines. The cylinders are supported at the front side by turned columns *O*. The condenser and the front columns are bolted to the soleplate *P* which in turn is bolted to the engine seating shown at *Q*, the seating being worked into the framing of the ship.

The **soleplate** has six main bearings, *R*, to receive the crank shaft *S*. In the sectional elevation, the mechanism connecting the intermediate pressure piston and slide valve to the crank shaft is shown complete. The connecting rods and valve gears of the other engines are omitted to reduce complication in the drawing.

The valves are driven, and the engines reversed and graded in expansion, by means of **Stephenson's link motions**, of which there are

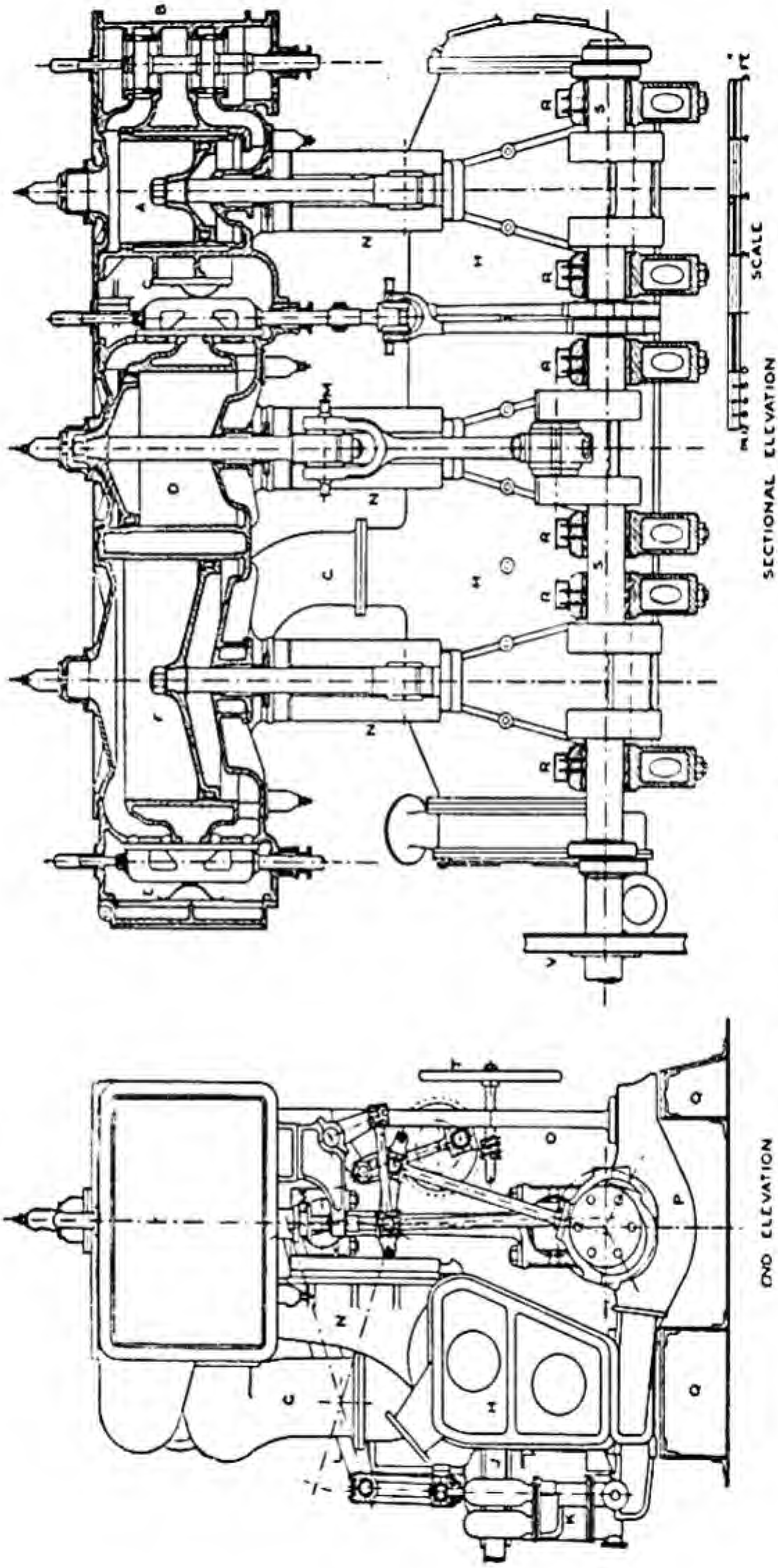


FIG. 202.—General arrangement of a set of triple expansion marine engines.

three sets. All the links are put over from ahead to astern simultaneously by operating the hand wheel *T*, a small steam reversing engine (not shown) being employed to assist the attendant. The engines may be moved slowly by means of a small turning engine, not shown in the illustrations, which is connected to a worm wheel *V* fixed to the crank shaft. The low pressure end of the crank shaft is nearest to the stern of the ship and is connected to a line of shafting carrying the propeller at its outer end.

Cylinders.—The principal points of information regarding the dimensions of the cylinders are given in the following Table:

	H.P.	I.M.P.	L.P.
Diameter of cylinder, - - -	21″	32″	50″
Stroke of piston, - - - -	24″	24″	24″
Diameter of piston rod, - - -	$4\frac{3}{4}$″	$4\frac{3}{4}$″	$4\frac{3}{4}$″
Ratio of areas of pistons, - -	1	2·18	5·68
Clearance volumes, as percentage of volume swept by pistons, -	23	11·25	8·1
Cut-off, as fraction of stroke, - -	·67	·67	·58
Total ratio of expansion, - -			8·45
Weight of reciprocating parts, lbs.,	1976	2256	2738
Travel of valve, - - - -	$4\frac{1}{2}$″	$4\frac{1}{2}$″	$4\frac{1}{2}$″
Lead on top, - - - - -	$\frac{3}{16}$″	$\frac{3}{16}$″	$\frac{3}{16}$″
Lead on bottom, - - - -	$\frac{7}{16}$″	$\frac{7}{16}$″	$\frac{1}{2}$″
Steam lap on top, - - - -	$1\frac{1}{4}$″	$1\frac{1}{4}$″	$1\frac{7}{16}$″
Steam lap on bottom, - - -	1″	1″	$1\frac{1}{8}$″
Exhaust lap on top, - - -	0	0	0
Exhaust lap on bottom, - - -	$\frac{1}{4}$″	$\frac{1}{4}$″	$\frac{3}{16}$″
Length of connecting rods, - -	4′ 6″	4′ 6″	4′ 6″

The engines are intended to give about 1350 I.H.P. at 145 revolutions per minute, with boiler pressure 160 lbs. per square inch by gauge.

The cylinders are formed out of three main castings, as may be seen by inspection of the general section in Fig. 202. Each cylinder with its valve chest forms one casting. Flanges are provided by means of which the cylinder castings are bolted securely together.

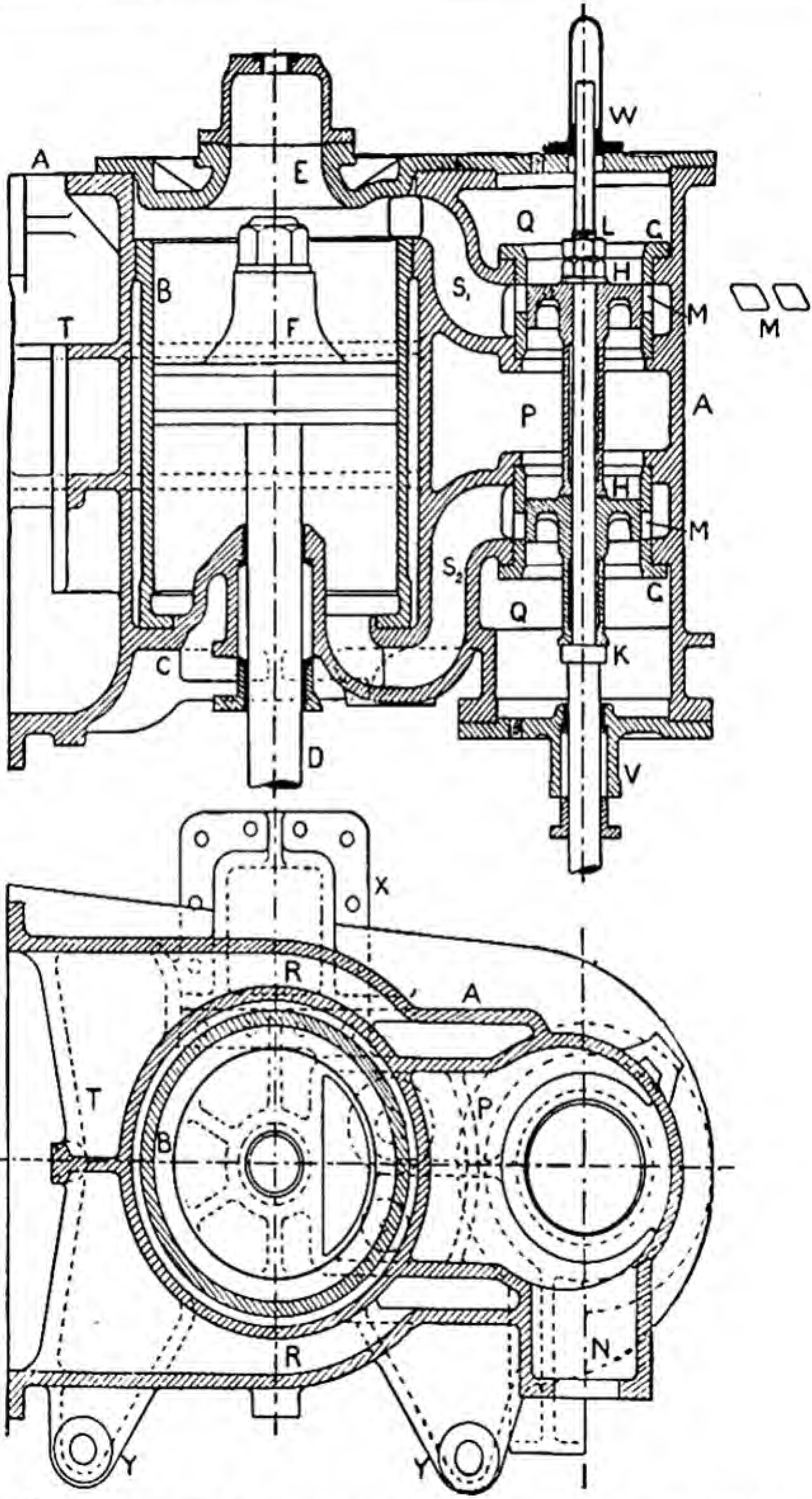


FIG. 203.—Sectional elevation and plan of the H.P. cylinder for a set of marine triple expansion engines.

High pressure cylinder details.—The high pressure cylinder is shown in sectional plan and elevation in Fig. 203. The main casting *A* is of cast-iron, and a cast-iron **liner** *B* is fitted to the barrel, the liner being flanged inwards at its lower end and secured there to the cylinder by means of screws. This connection is shown separately in Fig. 204. The liner should be made of hard close-grained metal so as to take a polish under the rubbing action of the piston. The lower cylinder cover *C* (Fig. 203) is cast in one piece with the main casting, and is provided with a stuffing-box and gland for rendering the piston rod *D* steam-tight. The upper

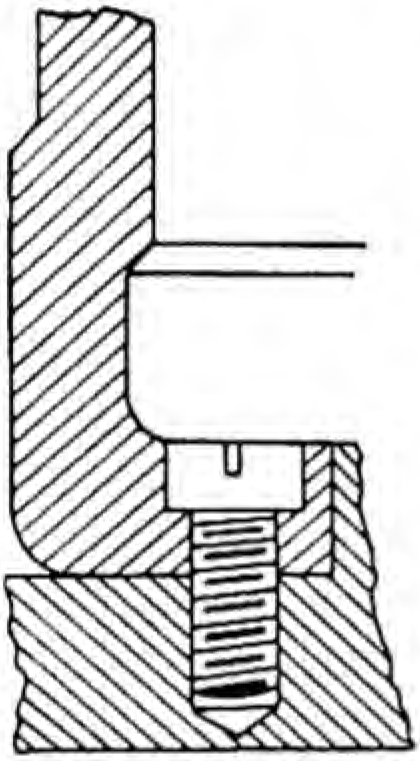


FIG. 204.—Method of securing the cylinder liner.

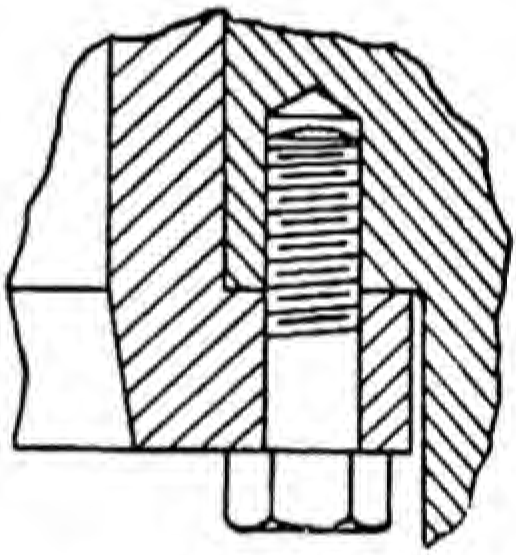


FIG. 205.—Method of securing the steam chest liners.

cover *E* is secured to the cylinder by means of studs. Both of these covers are made so as to suit the shape of the piston *F* (which is conical in form), and thus reduce clearance. The steam chest is bored out to receive cast-iron **liners** *G*, *G*, the liners being flanged as shown in Fig. 205, and secured to the main casting by means of screws passing through the flanges. The **piston valves** *H*, *H* (Fig. 203) are secured to the valve rod by means of a collar at *K* and lock nuts at *L*, distance pieces being also placed on the rod in order to have the valves kept at the proper distance apart. The piston valves are made of cast-iron, without packing rings, and move over the ports *M*, formed in the liner by casting a number of slots separated by diagonal bars. The object of having the bars arranged diagonally is to prevent the unequal wear which the

pistons of the valve would undergo with vertical bars. The ports M open into large passages S_1 and S_2 formed in the main casting, and leading one to each end of the cylinder. It will be noticed that the ports S_1 and S_2 are so arranged as not to interfere with the cylinder liner B. Steam enters the steam chest through the passage N into the space P between the piston valves, whence it is distributed to the cylinder. The exhaust steam escapes past the piston valves into the spaces Q, Q, which are connected by passages R, R leading round the cylinder into the space T, which forms part of the intermediate pressure valve chest.

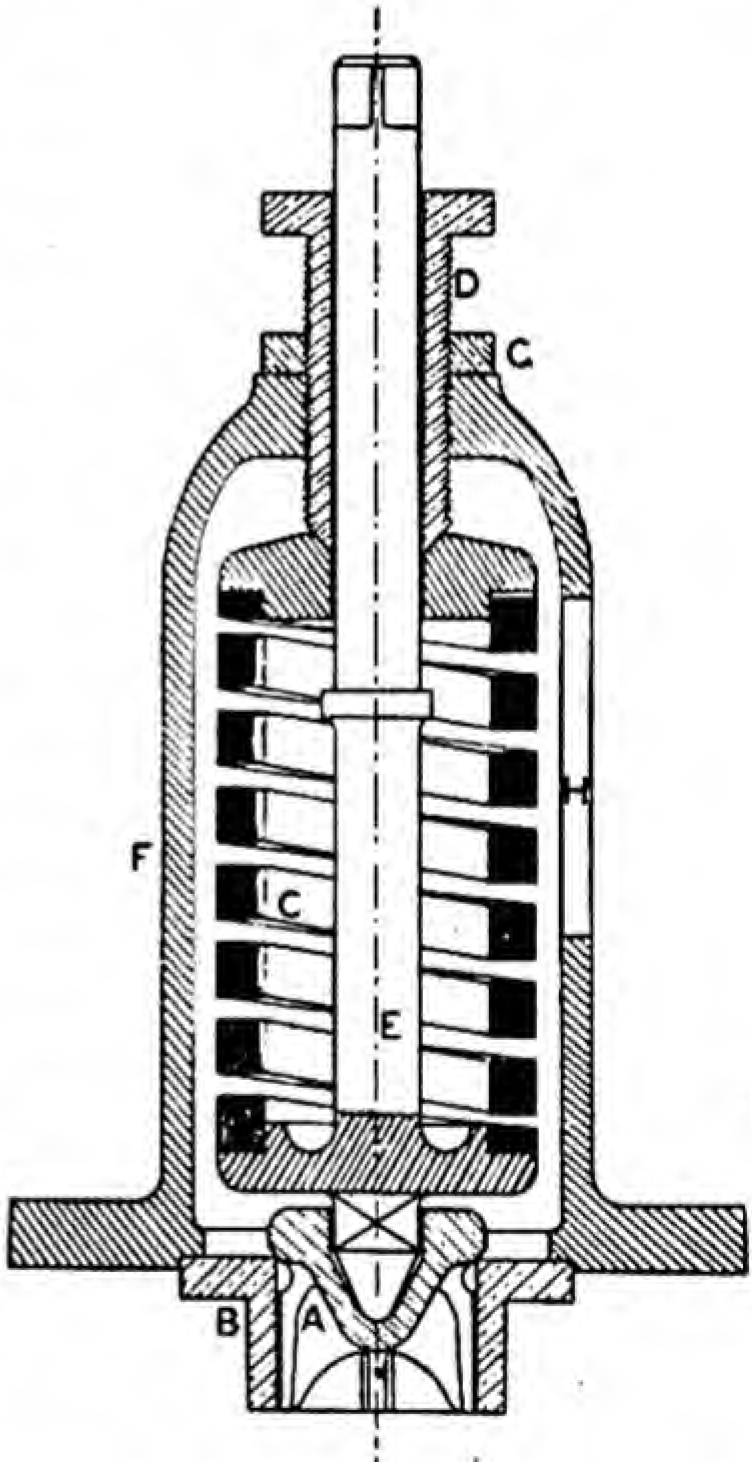


FIG. 206.—Marine cylinder relief valve.

The high pressure valve chest is fitted with top and bottom covers, secured to the main casting by means of studs. The bottom cover has a stuffing-box V, and the upper cover has a gun-metal bonnet W, which serves as a guide for the valve rod. A domed piece obviates the necessity for packing the rod in this place. The cylinder is fitted with the usual drain cocks and indicator pipes and cocks. Water accumulating in the cylinder, to an amount sufficient to cause damage to the covers or other parts of the cylinder, is when met by the returning piston got rid of through **relief valves.** One of these valves is provided for each end of cylinder, the valve being shown in detail in Fig. 206. A is the conical valve held down to its seat B by means of a strong helical spring C, the strength of which may be adjusted by means of the

screwed sleeve D which bears on the head of the spring. The lower end of the spring rests on a collar formed on a spindle E, the pointed end of which bears on the bottom of the recess of the valve A. The spindle has a square portion fitting a corresponding square part of the recess in the valve and has another square at its top end outside the domed case F. The object of the arrangement is to permit the valve to be rotated on its seat by means of a spanner applied to the square end of the spindle, and so to enable it to be freed periodically from sticking. The case F has a hole screwed to receive the sleeve D, which may be locked in position by the nut G. The case is secured to the cylinder by means of studs passing through the flange at its foot. The valve is loaded by the spring to such an extent that it remains closed during the normal working of the engine. Should the pressure inside the cylinder,

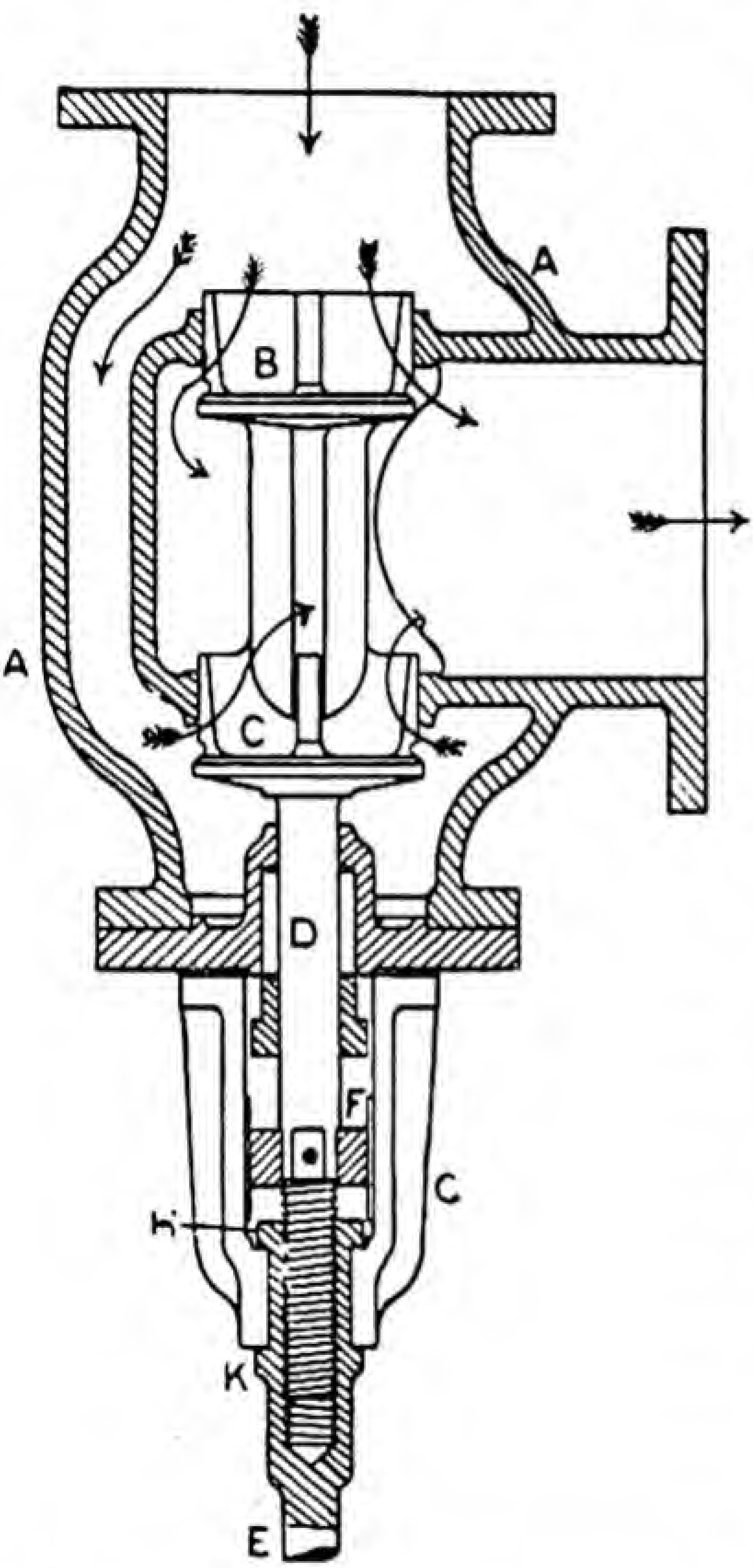


FIG. 207.—Marine equilibrium throttle valve.

due to the pressure of water, rise higher than that for which the relief valve is set, the valve will open and permit the water to escape through the hole H in the case G. The positions of the relief valves are shown in the general section in Fig. 202.

The high pressure cylinder (Fig. 203) has a bracket or foot cast to it at X for securing it to the back supporting column, and other

two feet at Y, Y, by means of which it is secured to the front columns. Provision is made for "lagging," or clothing, the cylinders by covering all outer surfaces with non-conducting material. Asbestos fibre, held in place by sheet iron or wood (teak or mahogany), is often employed.

Steam is admitted to the high pressure steam chest through a **throttle valve**, 6″ in diameter, of the equilibrium type. The valve is shown in section in Fig. 207. The case A has an outer and inner part with a passage between. The inner part has two valve seats to receive the valves B and C, which are of the conical type. The valves are in one piece with the spindle D, which is screwed to receive the operating spindle E, and has a guide block secured to it at F, working on prepared surfaces of the bridge G. The lower cover of the casing has a stuffing-box to receive the valve spindle. The operating spindle E has collars at H and K so that it may rotate without axial movement; the bridge is split to permit the spindle to be placed in position. Thus, rotation of the operating spindle will cause the valve spindle D to rise or fall and so to close or open the valves.

The valves are shown open in Fig. 207. It will be noticed that the steam presses on one side of the valve B and on the opposite side of the valve C. As both valves are nearly of the same diameter, it follows that these opposite forces nearly balance each other, and that very little effort will be required to open or close the valve.

Details of the other cylinders.—The intermediate pressure and low pressure cylinders are not fitted with liners. The arrangement of these cylinders is shown with sufficient clearness in Fig. 202. The slide valves are of the **double ported type**, the low pressure slide valve being shown separately in Fig. 208. The working of this type of valve has been already described in Chapter VIII., and the details of construction will be followed readily. The valve is secured to the valve rod by a collar below and locking nuts and split pin above, and is held up to its seat by means of coach springs, which rub on prepared surfaces formed on the steam chest casting.

Pistons.—The details of all three pistons are the same; one of them is shown in Fig. 209. The piston A is of cast steel of the **conical dished form**, this being well suited to resist without damage the great forces which the steam applies to it. The piston rod B

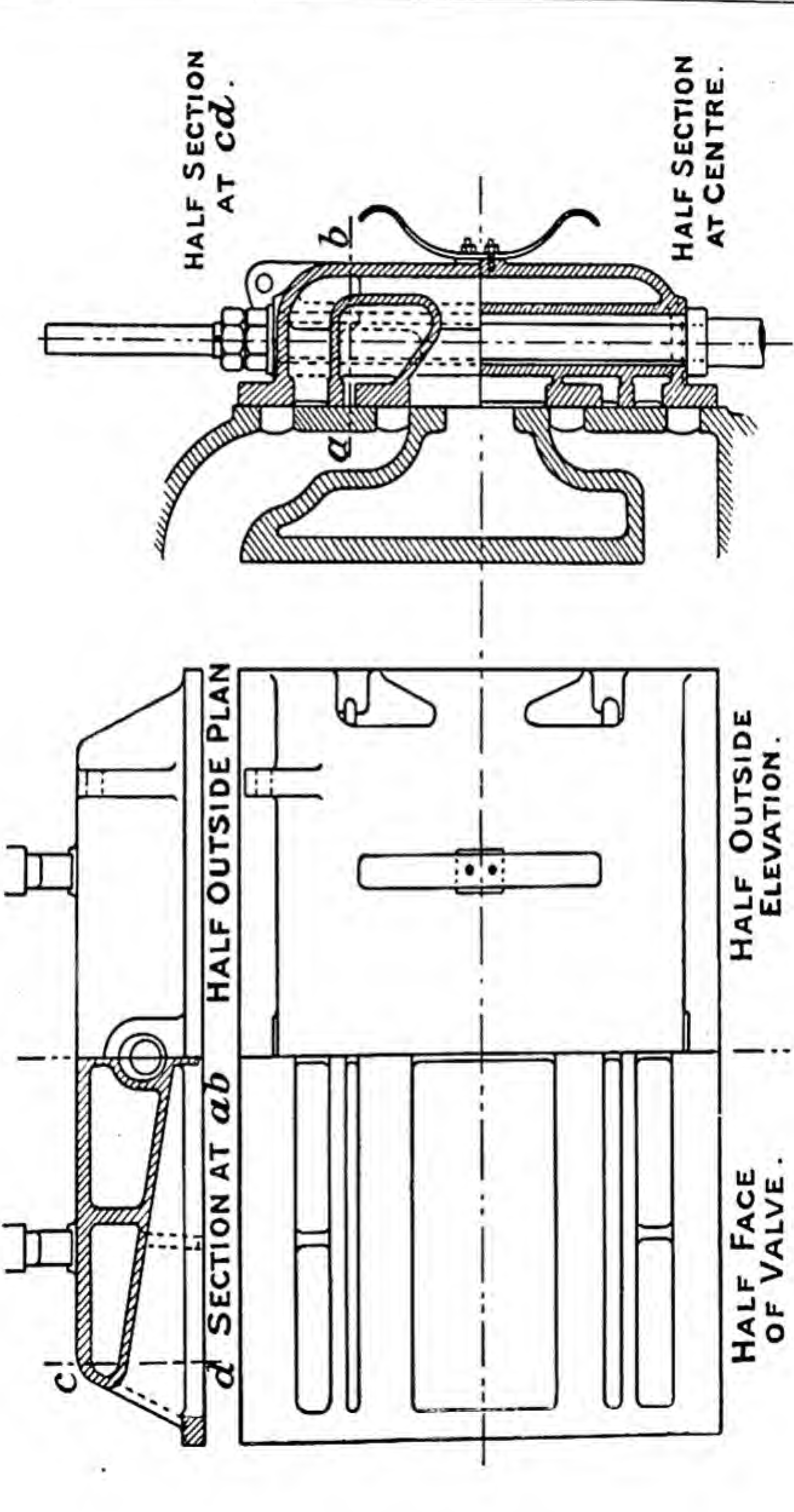


FIG. 208.—Double ported slide valve for the L.P. cylinder of a set of triple expansion marine engines.

is turned to fit the partly conical, partly parallel hole in the piston, and is secured by means of a $3\frac{3}{4}''$ Whitworth threaded nut *C*, made of gun-metal. The nut is capped to prevent any chemical action being produced between the metal of the rod and of the nut, which might occur if steam were permitted to get at the places of contact, and is locked by means of a $\frac{3}{8}''$ set screw. A snug on the tapered part of the rod prevents rotation of the piston on the rod.

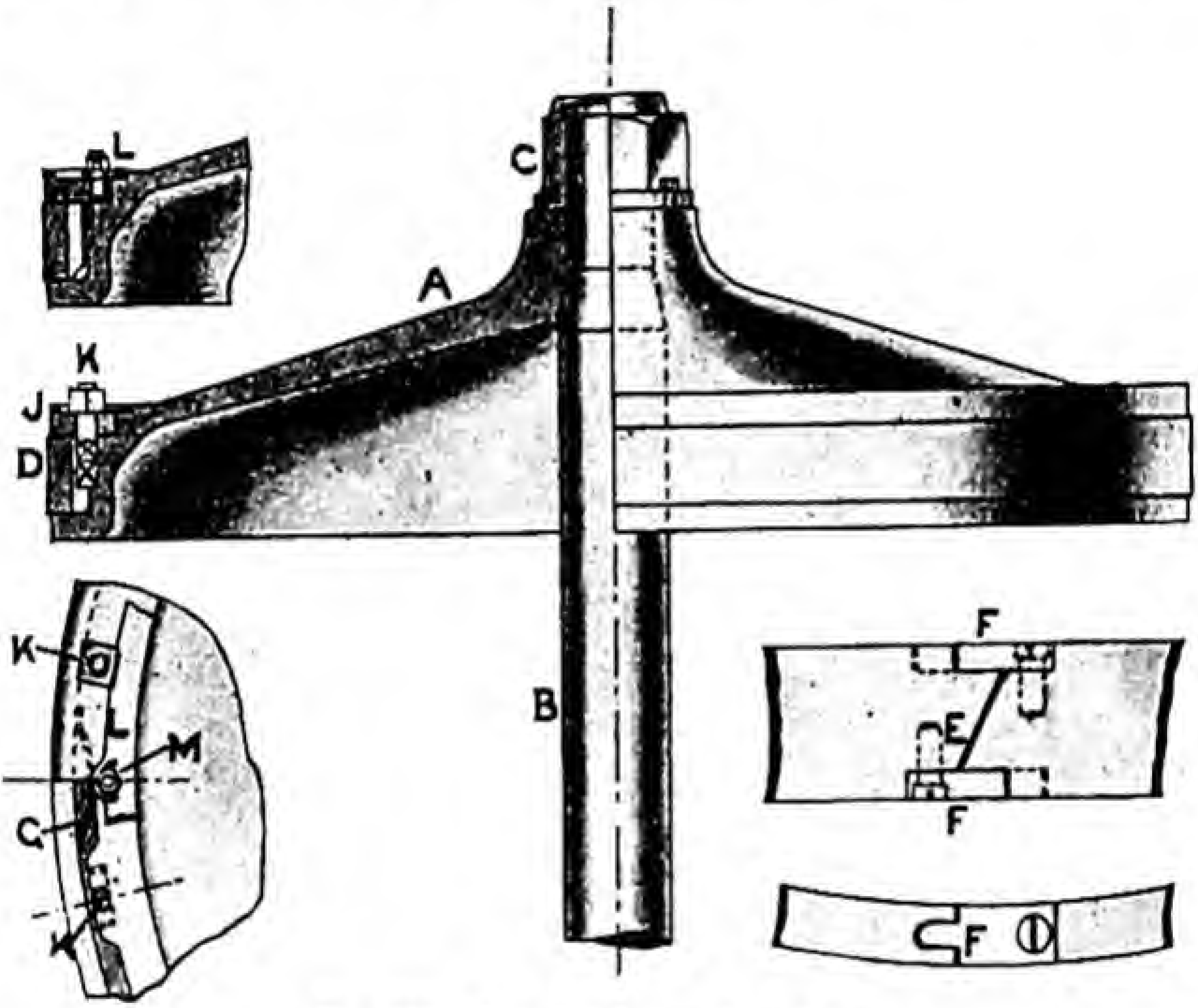


FIG. 209.—Details of a marine piston.

The rim of the piston is recessed to receive the cast-iron **piston ring** *D*, which is split, as shown separately at *E*, the split being covered by tongue pieces *F*, *F* secured by means of screws to the top and bottom edges of the ring, an arrangement which helps to prevent leakage of steam past the ring. The ring is pressed outwards against the cylinder walls by **coach springs** *G*, which are placed all round the recess. After the ring is placed in position in the recess, a **junk ring** *J* is placed over it to hold it in position. The junk ring is secured by means of steel T-headed bolts *K* with square brass nuts. These nuts are prevented from slacking back by means of a **locking ring** *L*, secured to the piston by a number of studs with brass nuts, one of which is shown at *M*.

These nuts are locked by split pins. It will be observed that great care is taken to prevent any of the piston fittings from becoming loose and so causing damage.

Crossheads.—The crossheads are all alike, consisting of rectangular blocks *A* (Fig. 210), forged to the lower ends of the piston rods and shaped to receive the crosshead brasses *B*. The brasses are held in position by means of a cap *C* and two bolts *D*, each

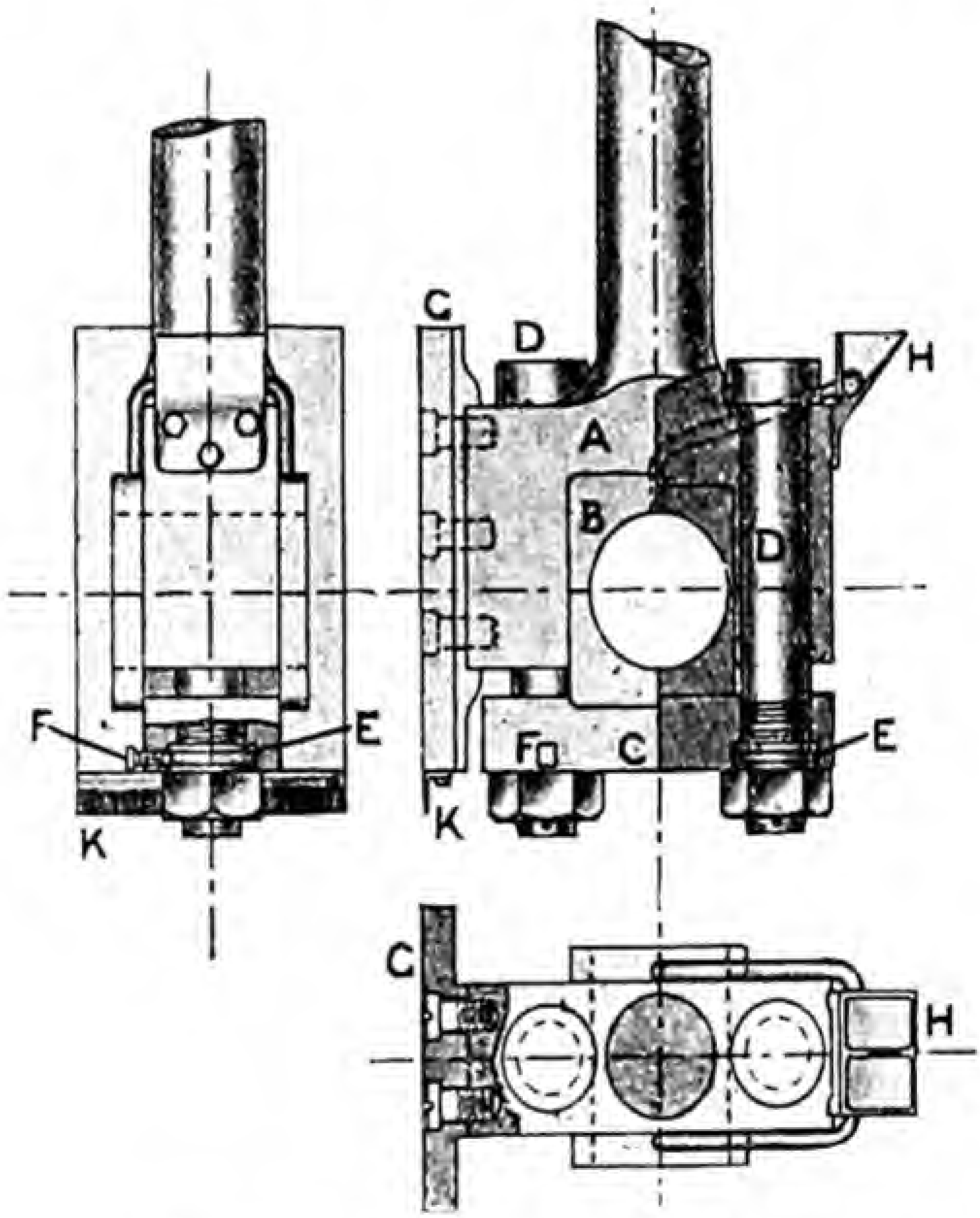


FIG. 210.—Marine crosshead and slipper.

$2\frac{3}{4}''$ diameter. To prevent the nuts slacking back, part of each nut is turned cylindrically to fit a corresponding recess *E* in the cap. A set screw *F* is inserted through a screwed hole in the side of the cap, its point bearing on a shallow groove turned in the cylindrical part of the nut. Split pins are also put through the points of the bolts as an additional safeguard. The **slipper** *G* is made of cast-iron, secured to the crosshead by means of cheese-headed screws Oil is fed to the crosshead bearing from the cup *H*.

Guides.—The **guides** consist of cast-iron plates *A* (Fig. 211),

bolted to the front sides of the columns, with **guide bars** *B* also fitted to embrace the slipper. The top bolts have collars at their middle part and nuts at both ends, thus permitting all the other bolts to be withdrawn and the guide bars adjusted without the guide plates being displaced. Waved channels are cut on the face of the guide plates to permit of a good distribution of the oil. An oil drip cup *C* is secured at the lower part of the guide to catch waste oil. A brass comb *K* (Fig. 210) is secured to the lower end of the slipper and serves to distribute the oil over the face of the guide plate.

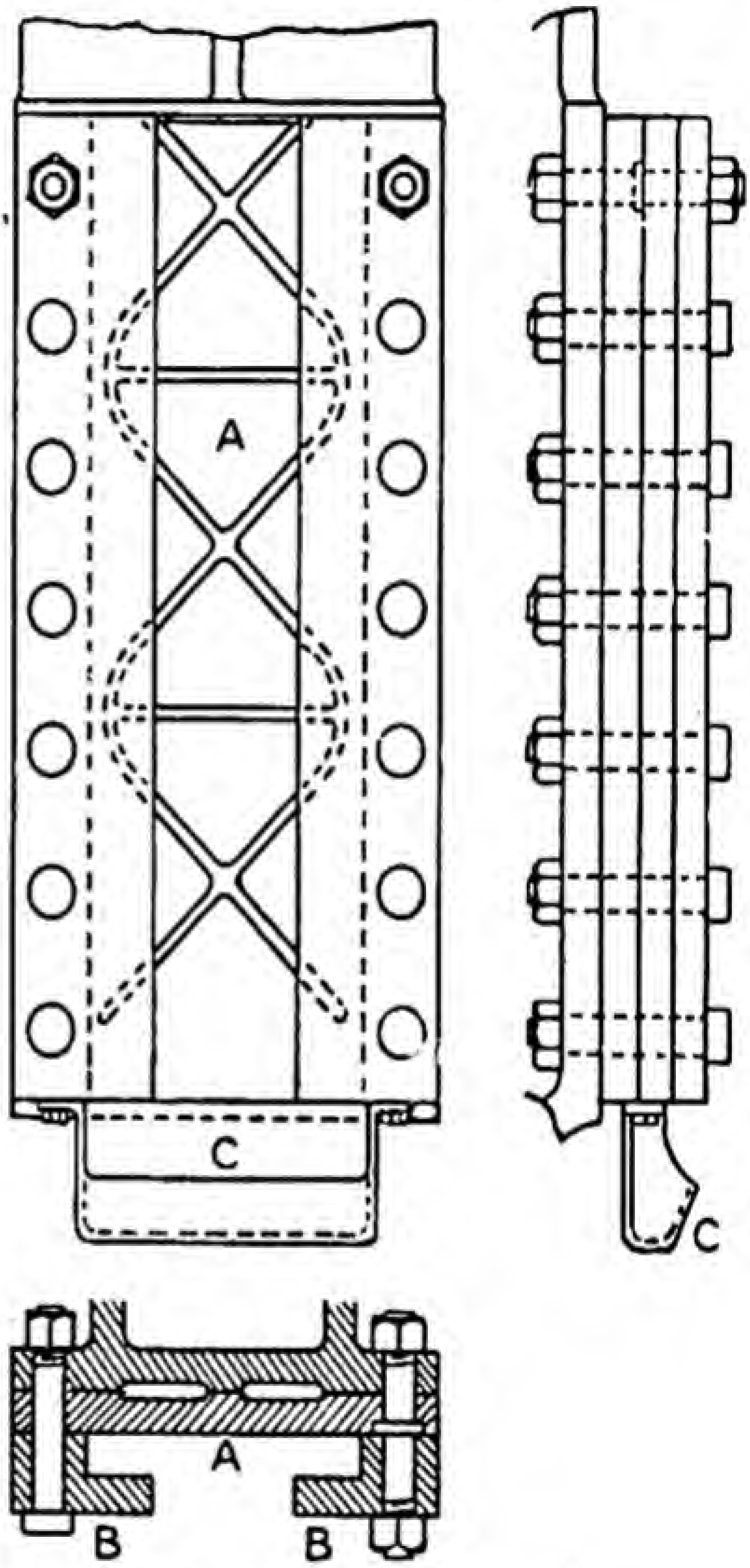


FIG. 211.—Marine guides.

Connecting rods.—These are all alike, and are jawed at the upper ends to embrace the crosshead and shaped at the lower ends to receive the crank pin brasses. In the intermediate rod shown in Fig. 212, the crosshead pin *A* is shrunk into place in the connecting rod, and has external journals *B*, *B* for driving the air pump levers. In the other rods these journals are not required. The crosshead pins are $5\frac{1}{2}''$ diameter and are case-hardened. The crank pin brasses consist of gun-metal bushes *C* lined with **anti-friction metal** *D*. Oil is supplied to the crank pin from two oil cups *E*, *E* through pipes *F*, *F* passing down the rod. The brasses are held in position by means of a cap *G* and two

bolts $2\frac{3}{4}''$ in diameter. The nuts are prevented from slacking back by means of a set screw *H*, screwed into a ring washer *K*, which

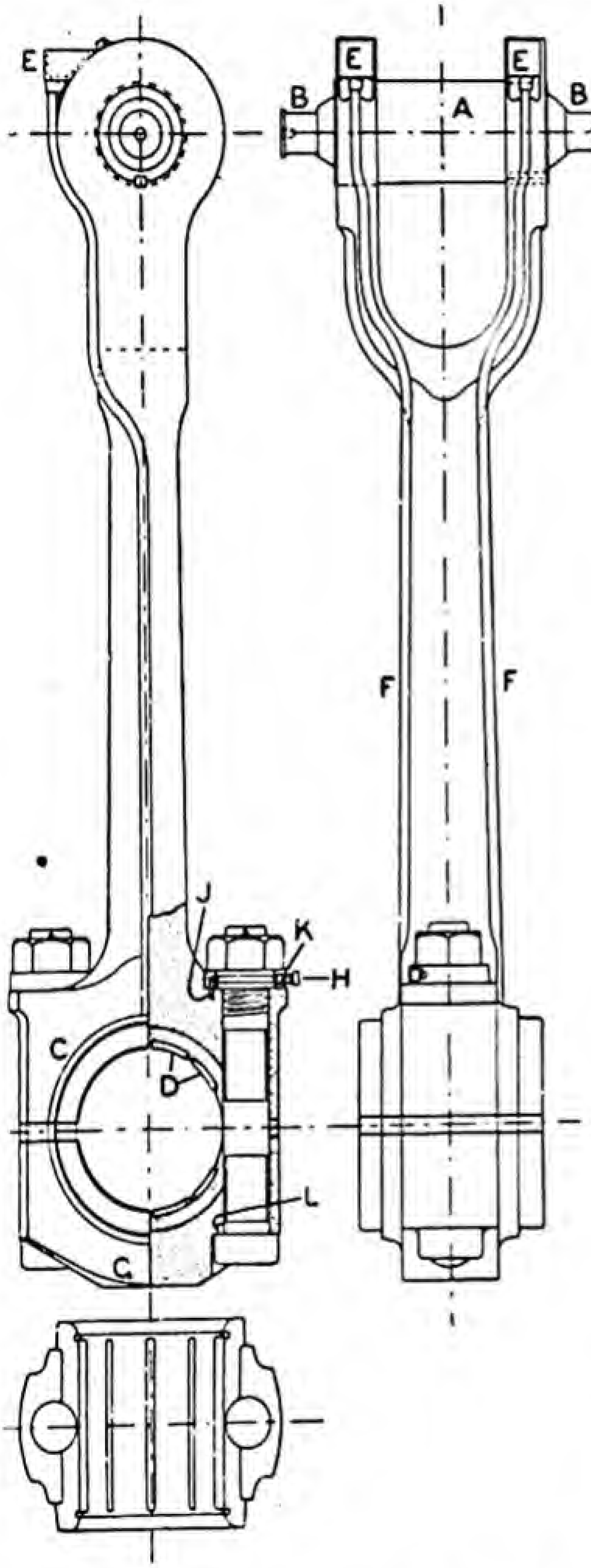


FIG. 212.—Marine connecting rod.

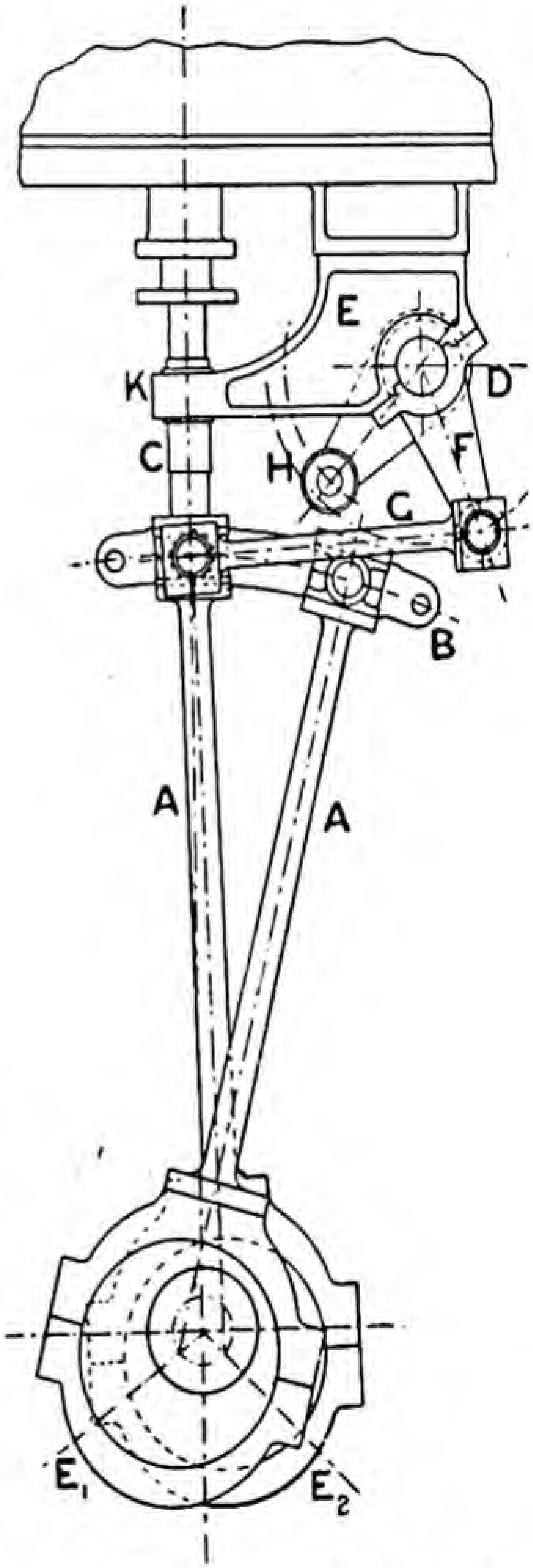


FIG. 213.—Valve gear for the set of marine triple expansion engines.

is prevented from rotation by means of a steady pin *J*. Part of the nut is made cylindrical to fit *K*. The point of the set screw

bears on a groove turned in this cylindrical portion. The bolts are prevented from rotation by means of snugs at *L*. The crank pin bearing is $8\frac{3}{4}''$ diameter. The connecting rods are $4\frac{1}{4}''$ diameter at the top and $4\frac{7}{8}''$ at the bottom, and are of best forged scrap iron. Piston rods in the type of engine under consideration are generally made of best forged ingot steel or highest quality forged iron.

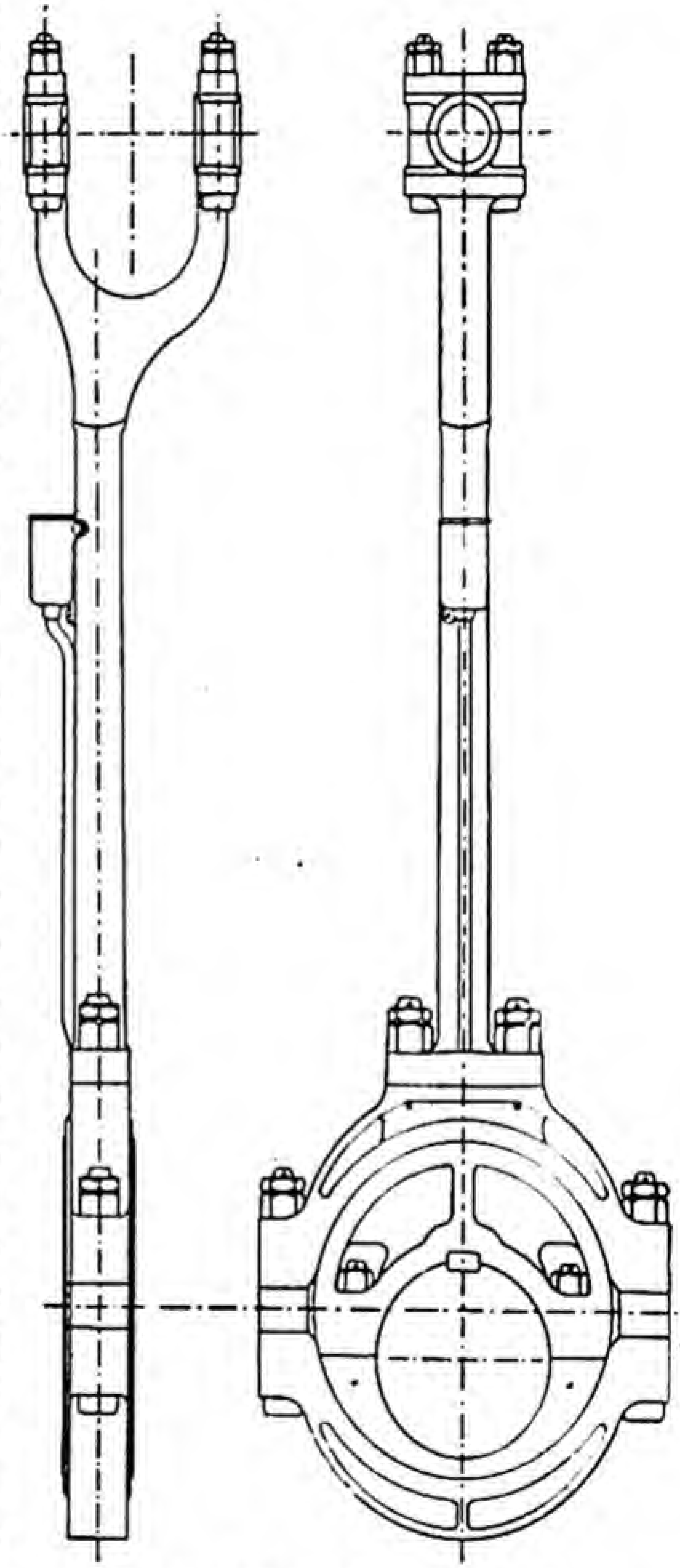


FIG. 214.—Marine eccentric and rod.

The valve motion.—The details of the Stephenson's link motion is the same for all three valves. The arrangement is shown in Fig. 213, where E_1 and E_2 are the eccentrics; *A*, *A* the eccentric rods; *B* the link and *C* the valve rod. A **reversing shaft**, *D*, is supported in bearings formed in brackets *E*, bolted to the under side of the cylinders, and has levers mounted on it, one of which is shown at *F*; these levers are connected to the three links by means of **drag links** *G*. On the lever *H*, also mounted on the reversing shaft, being operated from the hand wheel (*T*, Fig. 202), all three links are moved simultaneously. The bracket *E* is formed at *K* so as to serve for guiding the valve spindle.

Eccentrics and rods.—These are shown in detail in Fig. 214. The eccentric sheave is made in halves in order that it may be got into position on the shaft, the halves being secured together by studs and nuts locked by split pins. The thin lower half is made of wrought steel to give it sufficient strength. The strap is also in halves, the bolts being fitted with lock nuts. The eccentric rod is palmed at its lower end for attachment to

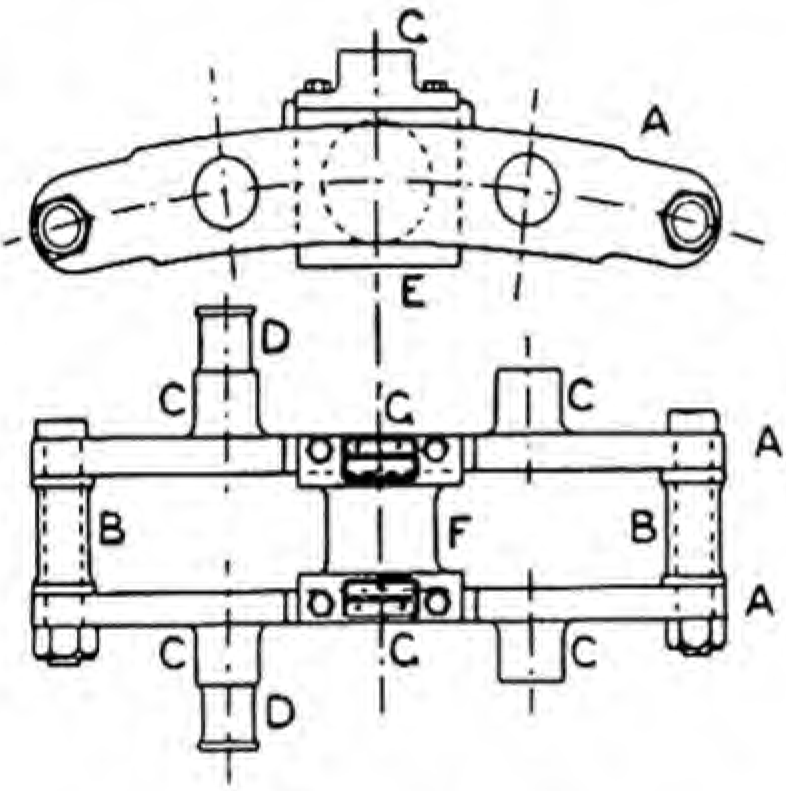


FIG. 215.—Marine link and block.

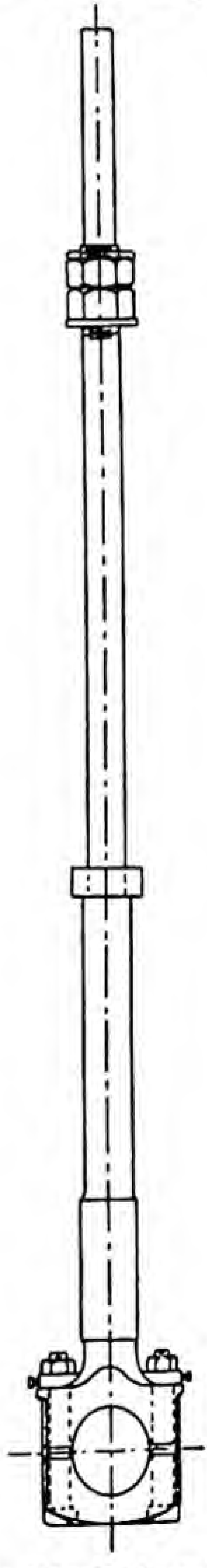


FIG. 216.—Marine valve rod.

the eccentric strap, studs having lock nuts and taper pins being used for the connection. An oil box is fixed to the side of the eccentric rod and a pipe is led from it to the eccentric for the supply of oil. The upper end of the eccentric rod is jawed and furnished with gun-metal bearings for receiving the link. The eccentrics have a crank radius of $2\frac{1}{4}''$, giving a maximum valve travel of $4\frac{1}{2}''$.

The link and block.—The **link** is double, as shown at *A*, *A* (Fig. 215), the pieces being held together by bolts and distance pieces *B*, *B*. **Journals** are forged at *C* to receive the bearings at the upper ends of the eccentric

rods. The journals *D*, *D* are for receiving the ends of the drag links, of which there are two for each motion, one on each side of the links. The **block** *E* slides between the links and embraces their top and bottom edges; the journal *F*, formed on the block, is for receiving the valve spindle. Oil is supplied to the rubbing surfaces of the block from two oil boxes *G*, *G*.

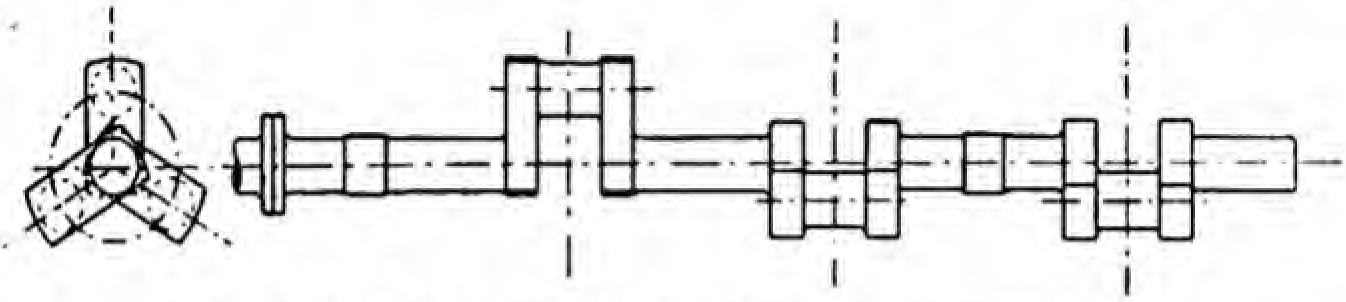


FIG. 217.—Triple expansion marine engine crank shaft.

The valve spindle is shown in detail in Fig. 216, and calls for no special description. It will be noticed that the method of locking the nuts of the adjustable bearing at the lower end is the same as that used in the main connecting rods.

The shafting.—The **crank shaft** is shown in detail in Fig. 217. It is made in one piece of forged steel, with cranks at 120°, and rests in six main bearings. It is $8\frac{1}{2}''$ in diameter at the bearings; the crank pins are $8\frac{3}{4}''$ in diameter. The shaft is connected to the **thrust** length of shafting by means of flanged couplings and bolts (Fig. 218). The flanges are forged solid on the shafts.

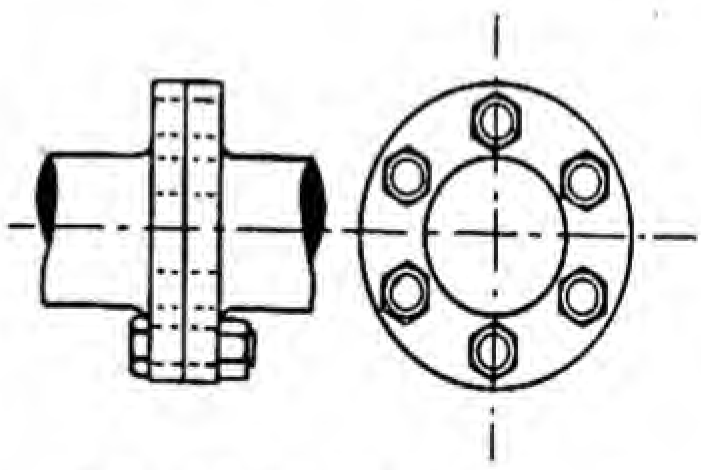


FIG. 218.—Flanged shaft coupling.

The general arrangement of shafting is shown in Fig. 219. *A* is the **crank shaft**; *B* the **thrust shaft**, having collars forged on it to receive the **thrust bearing**; *C* the **tunnel shaft** which is connected to the **tail shaft** *D*, carrying the propeller *E* at its outer end. Couplings similar to that shown in Fig. 218 connect these various shafts. The tunnel shaft runs in bearings at *F*, *F*. The tail shaft runs for the greater part of its length, sheathed with gun-metal, in a **stern tube** *A* (Fig. 220). The stern tube is secured to the stern post *B* and to a bulkhead at *C*, and is fitted with lignum vitae

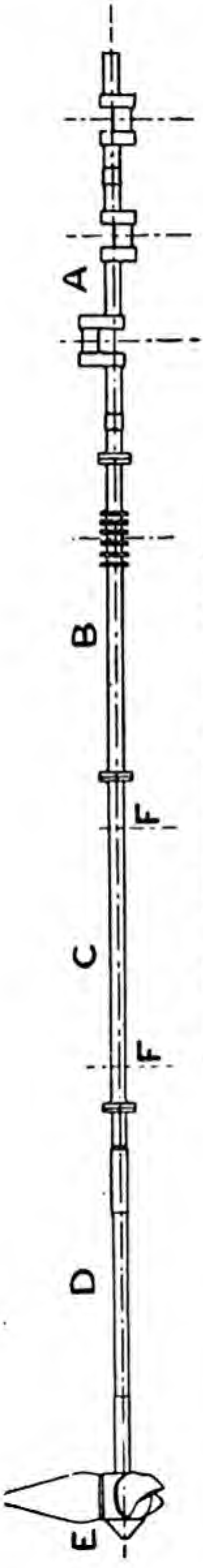


FIG. 219.—Arrangement of marine shafting.

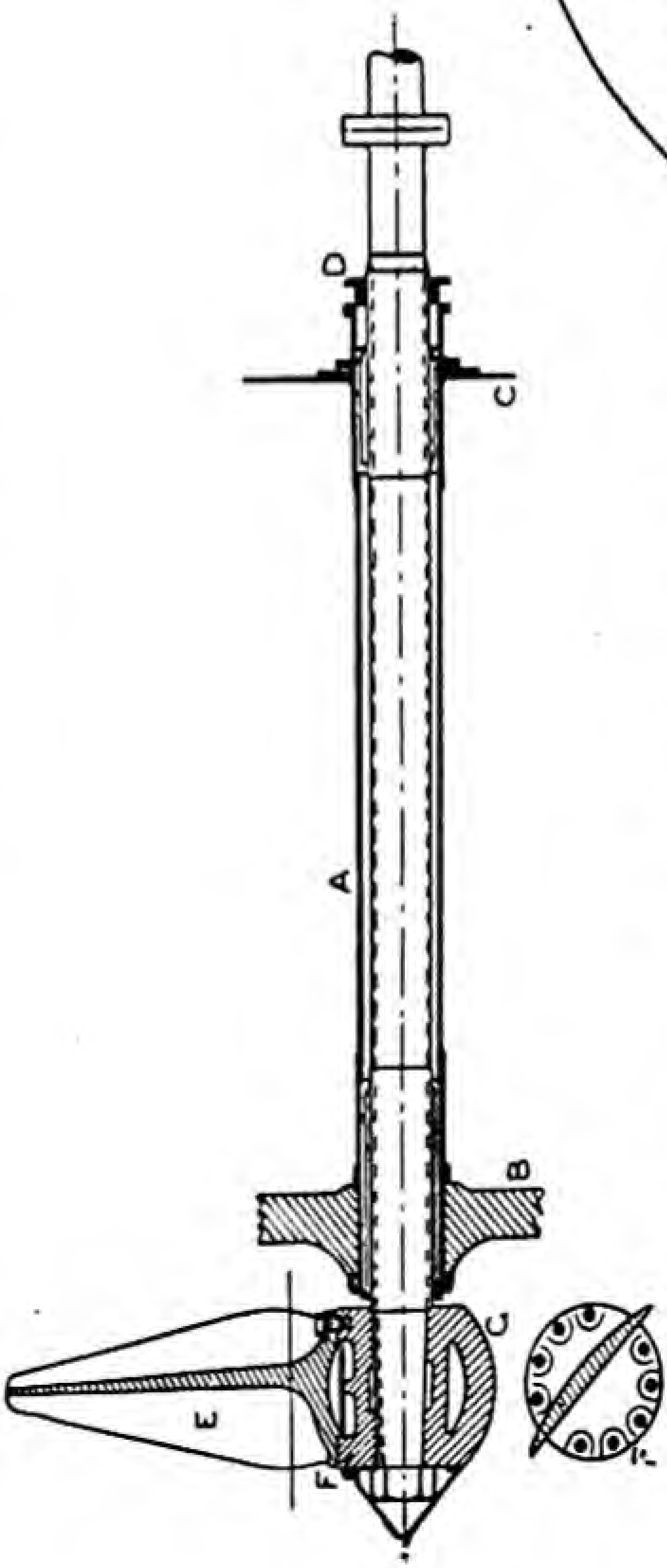


FIG. 220.—Tail shaft, stern tube and propeller.

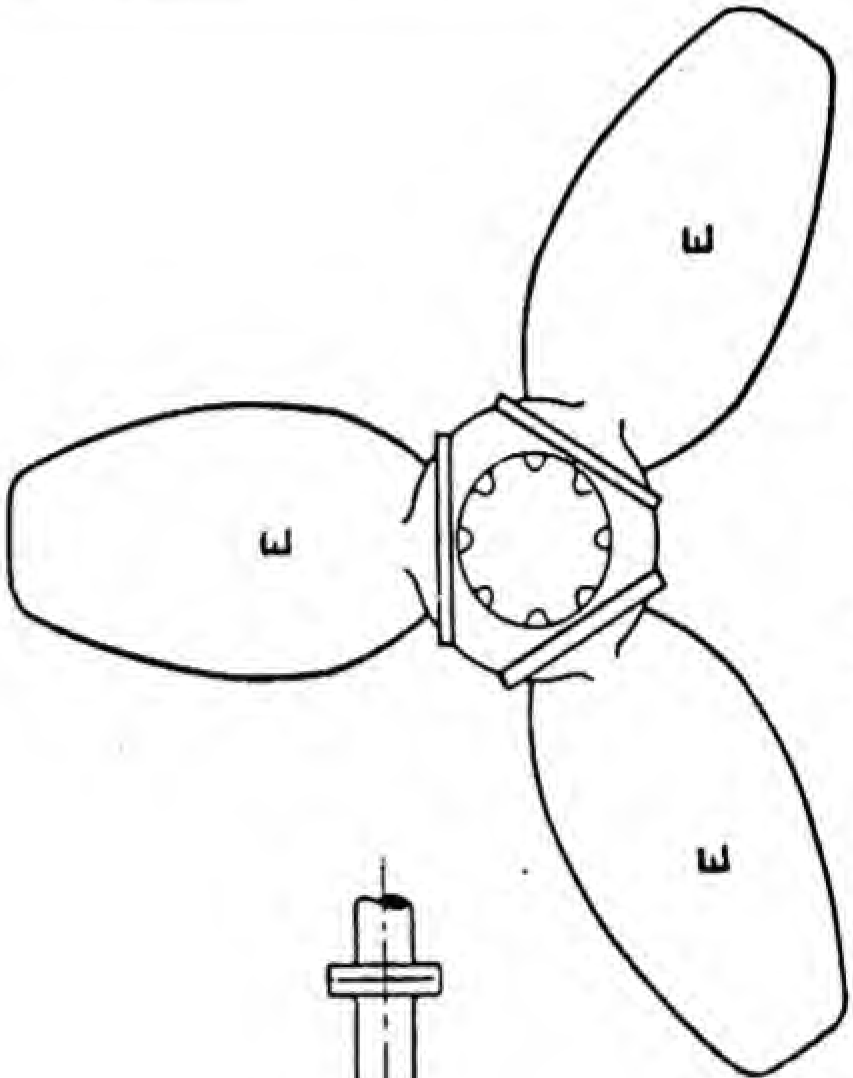


FIG. 221.—End elevation of propeller.

bearings and a stuffing-box at *D* to prevent leakage of sea water into the shaft tunnel.

The propeller.—The propeller is of the three-bladed type, each steel blade *E* (Figs. 220 and 221) being cast separately with a flange *F* for connecting it to the cast-iron boss *G*, to which it is secured by steel studs, the gun-metal nuts being locked by set screws. The boss is bored to fit the tapered end of the tail shaft, and is held in position by a large nut screwed to the shaft. A feather $1\frac{3}{4}'' \times 1''$ on the tapered end of the tail shaft prevents rotation of the propeller on the shaft. To keep sea water from the nut and also to taper off the boss easily, a conical cast-iron cover is placed over the nut and is secured to the boss by steel set screws. It will be observed that the blades have sharp edges, and are flat on the side remote from the ship and fish back on the other side. The propeller is 10′ 6″ in diameter × 10′ 6″ pitch and is right-handed.

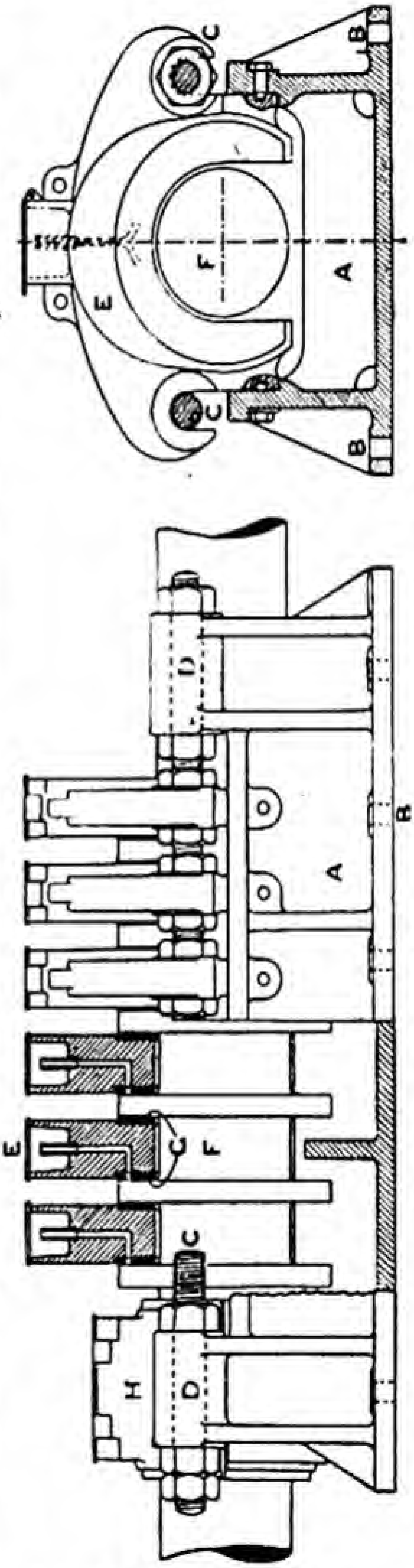


Fig. 222.—Sectional side and end elevations of a thrust block.

The **action of the propeller** consists in driving a stream of water astern. The inertia of the water causes a reaction, which gives a forward thrust to the propeller, which in turn is communicated to the shaft and has to be transmitted to the hull of the ship by means of the thrust block.

The thrust block is shown in detail in Fig. 222. It consists of a large casting *A*, strongly secured to the frame of the ship

by bolts passing through the holes at *B*. Two screwed stays, *C*, *C*, are attached to the casting *A* at *D*, *D* by means of nuts and hold the six **horse-shoe bearings** *E*. These bearings embrace the grooves between the collars on the thrust shaft *F*, the collars being in contact with the bearings on the sides only, these being faced with antifriction metal at *G*. The horse-shoes are adjustable on the stays *C* by means of nuts placed on the stays on each side of each horse-shoe. There is an ordinary shaft bearing formed in the thrust block at *H* to centralise the shaft. Each horse-shoe has an oil box at the top with channels leading to the rubbing surfaces. It will be noticed that the thrust coming along the shaft is taken by the horse-shoe bearings, and thus transmitted through the block to the hull of the ship.

The soleplate.—The soleplate consists of a large casting in iron (Fig. 223). There are six **main bearings** to receive the crank shaft, one being shown in section in the end view. The bearings are of gun-metal with antifriction metal linings, and are held down by cast-iron caps and long bolts passing through the soleplate. Locking arrangements similar to those used in the connecting rods are provided for the cap nuts. Split pins are provided for both top and bottom nuts. Planed strips are provided on the top surface of the soleplate to receive the columns, condenser, air and feed pumps. Gaps are left between the main bearings for the cranks to work in ; these are called **crank pits**.

The condenser.—The surface condenser is shown in section in Figs. 224 and 228. It consists of a large iron body casting *A* containing 700 brass tubes running at *D* between rolled brass **tube plates** *B* and *C*. The tubes are $\frac{3}{4}''$ external diameter $\times$ 18 B.W.G. in thickness. The tube plates are $\frac{7}{8}''$ thick, the distance between them being 12′ 9″. The **water ends** of the condenser are formed of separate iron castings bolted to the body. The water end *E* is provided with a circulating water inlet *H*, a baffle plate *F* and a circulating water outlet *K*. Water entering *H* is thus compelled to flow through the lower rows of tubes to the other water end *G*, then back through the upper tubes into *E* and so discharged. Inspection doors are placed at *O* in the ends and body. The steam to be condensed enters the condenser at *L* from the low pressure cylinder and is distributed throughout the tubes by the baffle plate *M*, which is of brass with a number of 1″ holes punched in it. The

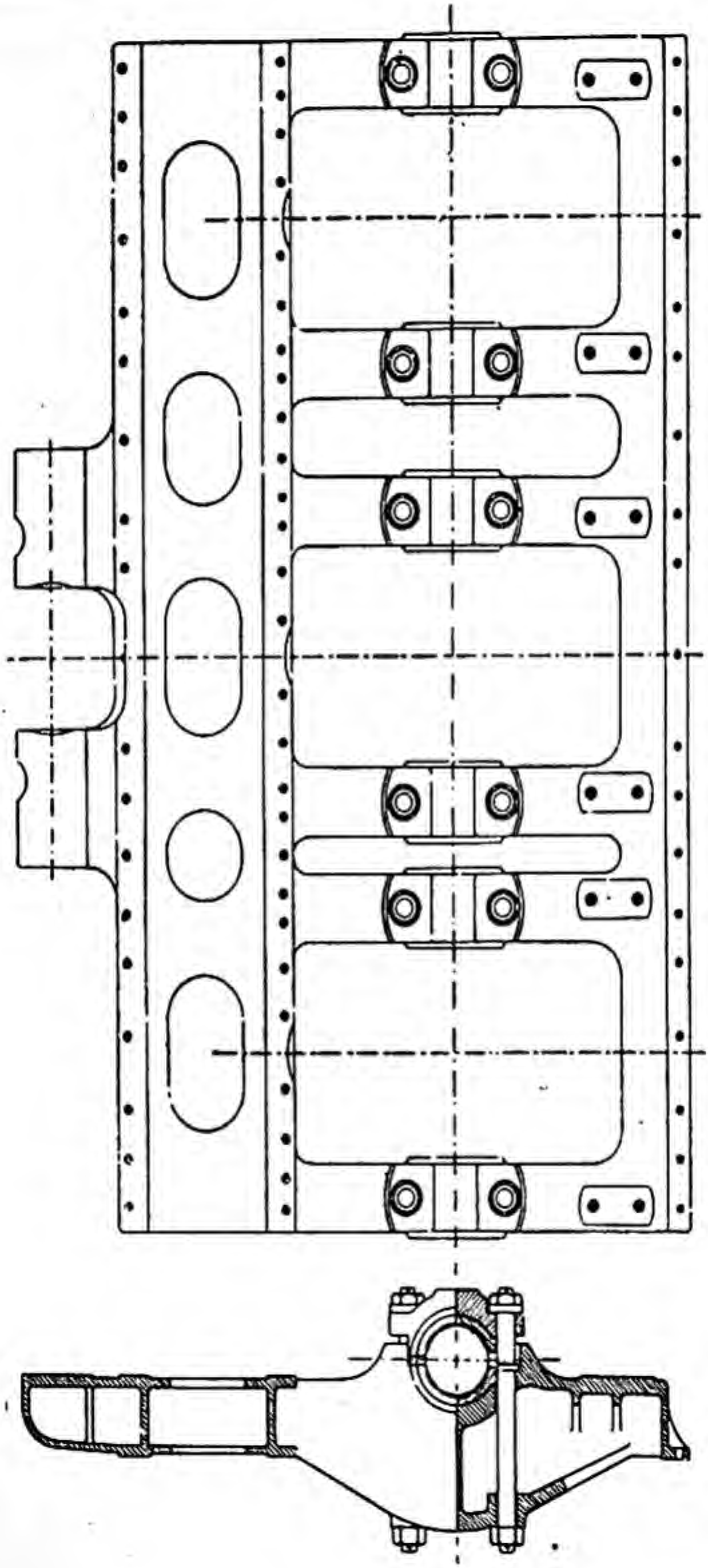


FIG. 223.—Soleplate and main bearings.

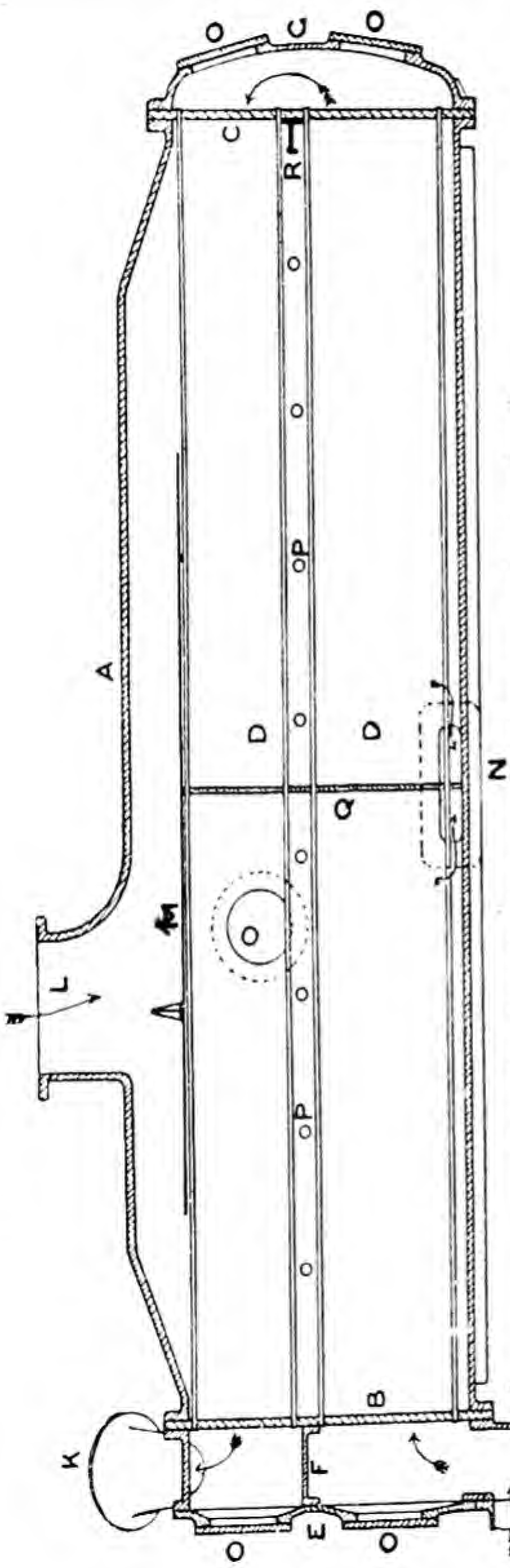


FIG. 224.—Longitudinal section of marine surface condenser.

resulting water is discharged to the air pump through *N* at the bottom of the condenser. A supporting plate *Q*, placed near the middle of the condenser, has holes drilled to receive the tubes and supports part of their weight. The large flat sides of the condenser are stayed by means of stiff stays *P*. It will be remembered that it is a collapsing pressure which has to be provided against. The tube plate *B* is sufficiently stiffened by the baffle plate *F*, the other tube plate *C* has a stiffening bar *R* secured to it. The condenser is tested for strength by subjecting it to an internal steam pressure of 30 lbs. per square inch.

The tube plates and ends are bolted to the body by means of collar bolts shown in detail in Fig. 225. The arrangement permits of the ends being removed without disturbing the tube plates. The tubes are spaced as shown in Fig. 226. To prevent leakage from the ends into the body, each tube passes through stuff-

ing boxes in the tube plates. The arrangement is shown in section in Fig. 227. The gland for pushing the packing down consists of a ferrule screwed into the box and made so that the tube cannot be drawn without first removing the ferrule. Two slots at the outer end of the ferrules permit of a tool being used for screwing them home. The cooling surface amounts to 1753 square feet.

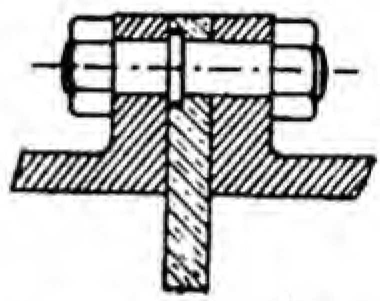


FIG. 225.—Condenser tube plate joint.

It is important that sea water should be kept out of the body of the condenser, where it would mix with the water of condensation and be fed into the boiler. If no losses whatever occurred, the condensation water from the condenser would be sufficient to feed the boilers. To make losses good, however, additional make-up water must be fed to the boiler. This is accomplished by means of

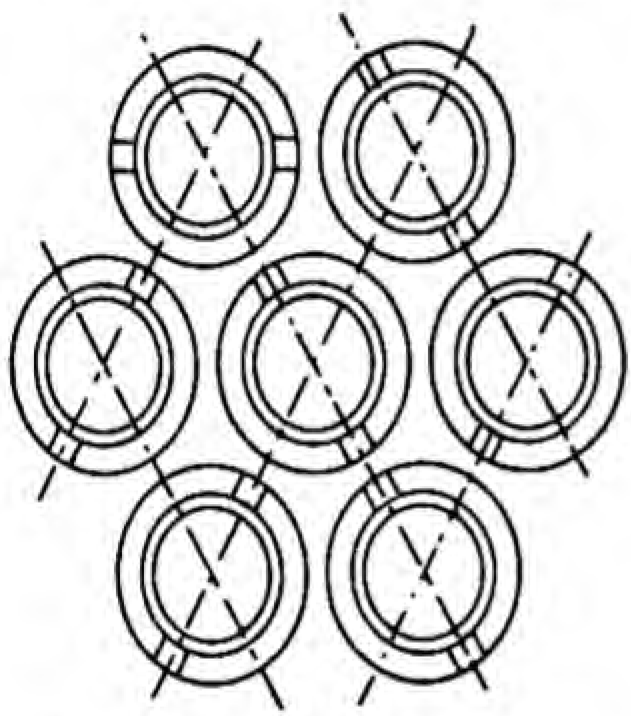


FIG. 226.—Condenser tube spacing.

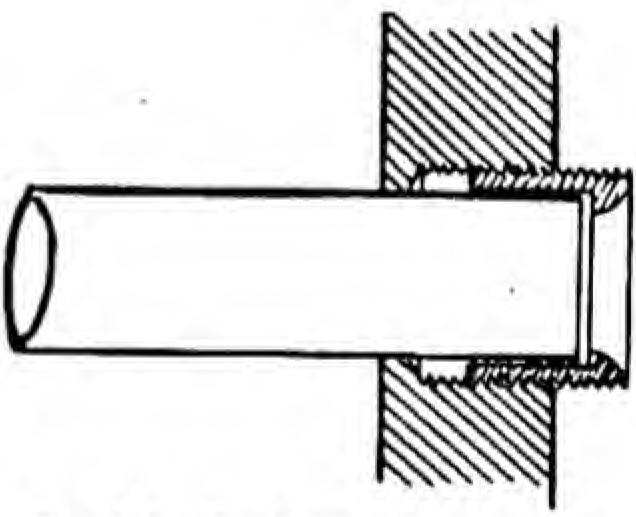


FIG. 227.—Condenser tube stuffing-box and gland.

an evaporating arrangement, which operates by distilling sea water, the resulting steam being discharged into the condenser. Normandy's Patent distilling machinery was fitted to this vessel and was capable of evaporating $7\frac{1}{2}$ tons of sea water per 24 hours. For use in case of emergency a by-pass is provided from the forward water end of the condenser to the body. By opening a cock, sea water may flow from this end into the body and so may be used as an auxiliary feed.

Solid matter in feed water.—Sea water contains about $\frac{1}{32}$ of its weight of solid matter in solution. If such water were evaporated continuously in the boilers, all the solid matter being left behind in the boilers would presently bring about unworkable conditions. It therefore becomes necessary to test the boiler water daily during running in order to ascertain that dangerous proportions of solid matter have not been reached. A **hydrometer** or **salinometer** is used, which, on being floated in a bucket of water drawn from the boiler, indicates on its scale the proportion of solid matter present. The naval hydrometer is marked in degrees and marks 0° when floated in fresh water and 10° in sea water containing 5 ounces of solid matter per gallon. It is not customary to exceed 40°, *i.e.* 20 ounces per gallon. It is not advisable to have the boiler water too fresh as this leads to "bleeding" (rusting) at the joints of the boiler. Should the proportion of solid matter exceed that above stated, the remedy consists in blowing off some of the water from the boiler and making up with feed water so as to reduce the density.

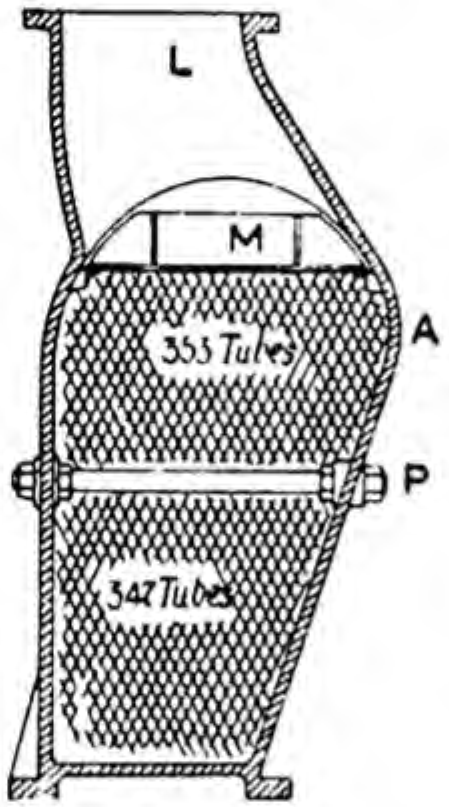


FIG. 228.—Cross section of marine surface condenser.

The air pump.—The **air pump** is shown in Fig. 229. It consists of a large brass body casting *A* bolted down to the soleplate *B* and to the condenser discharge *C*. The **bucket** *D* is of brass, secured to the pump rod *E* by means of a brass nut secured by a split pin, and rendered tight against leakage by means of a gun-metal spring ring *F*. The **bucket valves** are shown at *G*. *H* is a brass plate secured to the bottom of the pump by a central brass bolt and fitted with a number of **foot** or **suction valves** *J*. *K* is another brass plate held in position by the top cover *L*, which is bolted down to the cast-iron top of the pump *M*, this portion forming the **hot well.** *K* is fitted with a number of **top** or **delivery valves** *N*. The pump operates thus. During the down stroke the foot valves *H* and the delivery valves *N* remain closed. Any water and air present under the pump bucket will find their way through the

bucket valves *G* to the top side of the bucket. On the up stroke, the bucket valves *F* will be closed since the atmospheric pressure outside is greater than the condenser pressure, and the water and air on the top side of the bucket will be discharged through the delivery valves. At the same time, the pressure in the space between the bucket and the foot valves will be lowered, and presently the foot valves will open, allowing more water and air to pass through from the condenser. The pump is single-acting, delivering air and water on the up stroke only.

The **pump rod** is rendered air- and water-tight by a stuffing-box and gland in the top cover, and is guided by a slipper *P* attached to the pump crosshead *Q*, the slipper sliding on a rod *R*, which is secured to the air pump and to the side of the condenser. The pump bucket is 17″ in diameter × 12″ stroke.

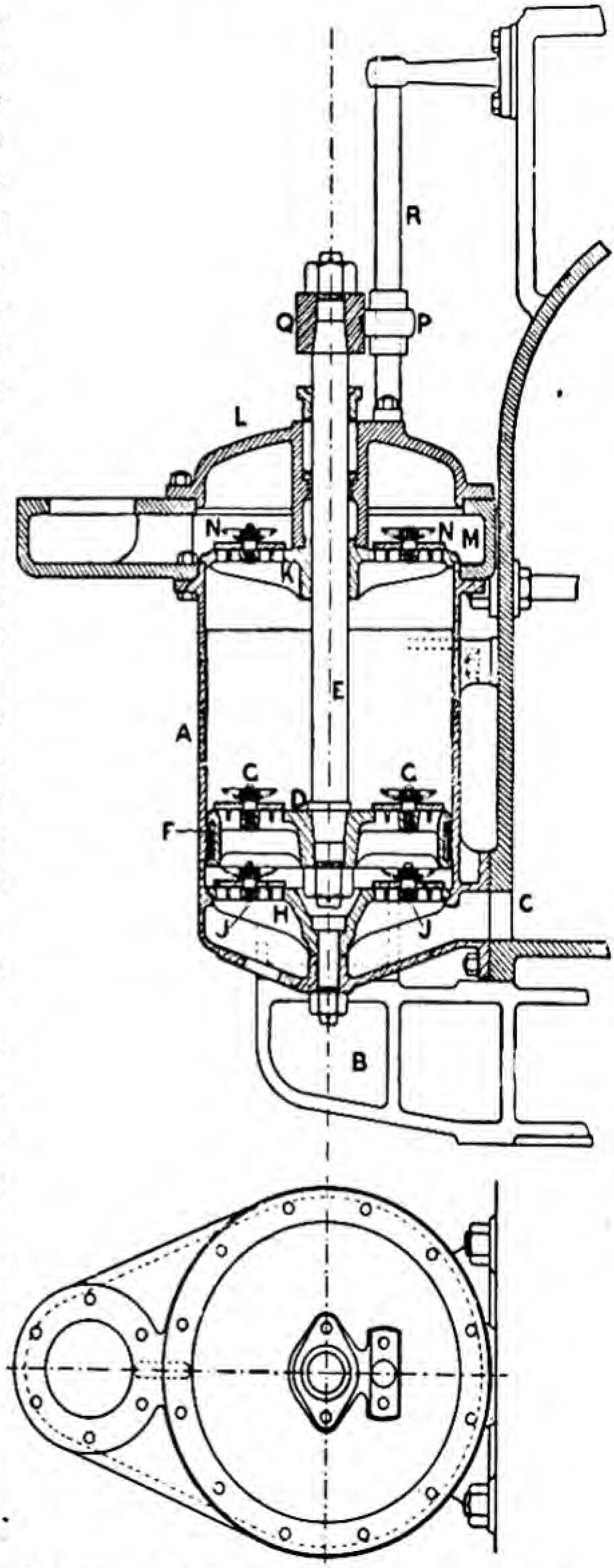


Fig. 229.—Sectional elevation and plan of marine air pump.

The **valves** are shown separately in Fig. 230. The valve *A* consists of a rubber disc covering a grating *B*, and is prevented from bending upwards too much by means of a guard plate *C*. A screwed stud *D*, of naval brass, holds all in position, the stud being screwed into the grating and riveted over at its lower end; the guard plate is screwed on to the stud and locked by a split pin. As rubber deteriorates rapidly, metallic valves are used extensively in modern practice.

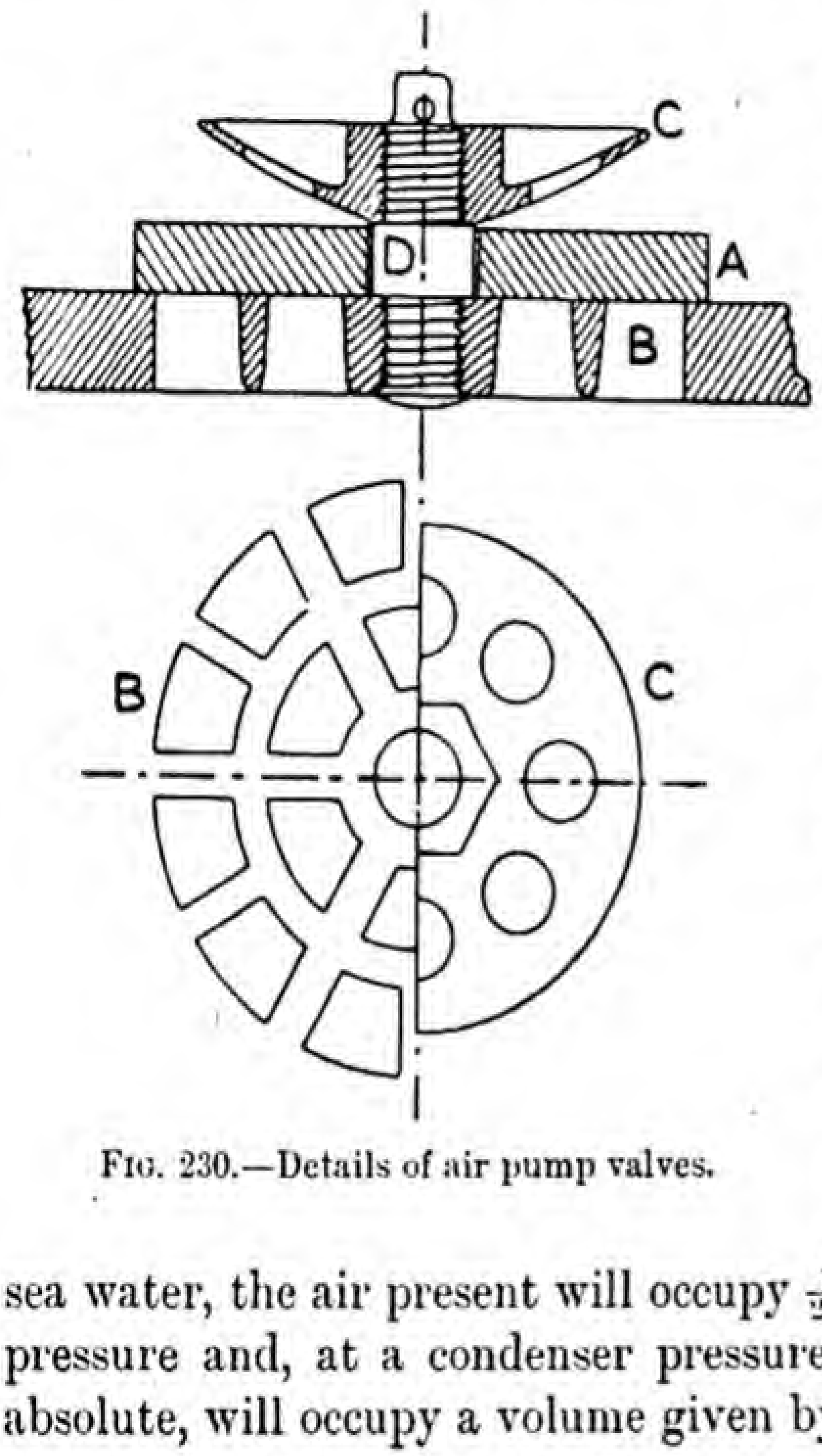


FIG. 230.—Details of air pump valves.

Air in feed water.—At ordinary atmospheric pressure, sea water contains air in the proportions of about 1 to 20 by volume. This air is mechanically mixed with the water and is liberated in the boiler and carried with the steam through the engine and so into the condenser. Considering one cubic foot of sea water, the air present will occupy $\frac{1}{20}$ cubic foot at atmospheric pressure and, at a condenser pressure of 2 lbs. per square inch absolute, will occupy a volume given by

$$\text{volume} = \frac{1}{20} \times \frac{15}{2},$$
$$= \underline{\frac{3}{8}} \text{ cubic foot.}$$

The air pump must therefore be capable of removing all the water of condensation from the condenser and, in addition, $\frac{3}{8}$ cubic foot of air for each cubic foot of sea water fed into the boiler.

The feed pump.—The feed pump for forcing the water into the boiler is shown in section in Fig. 231. It consists of a gun-metal body *A* fitted with a brass plunger *B* made hollow for lightness. The plunger is 3″ in diameter by 12″ stroke and there are two such pumps fitted. The plunger is rendered tight against leakage by means of

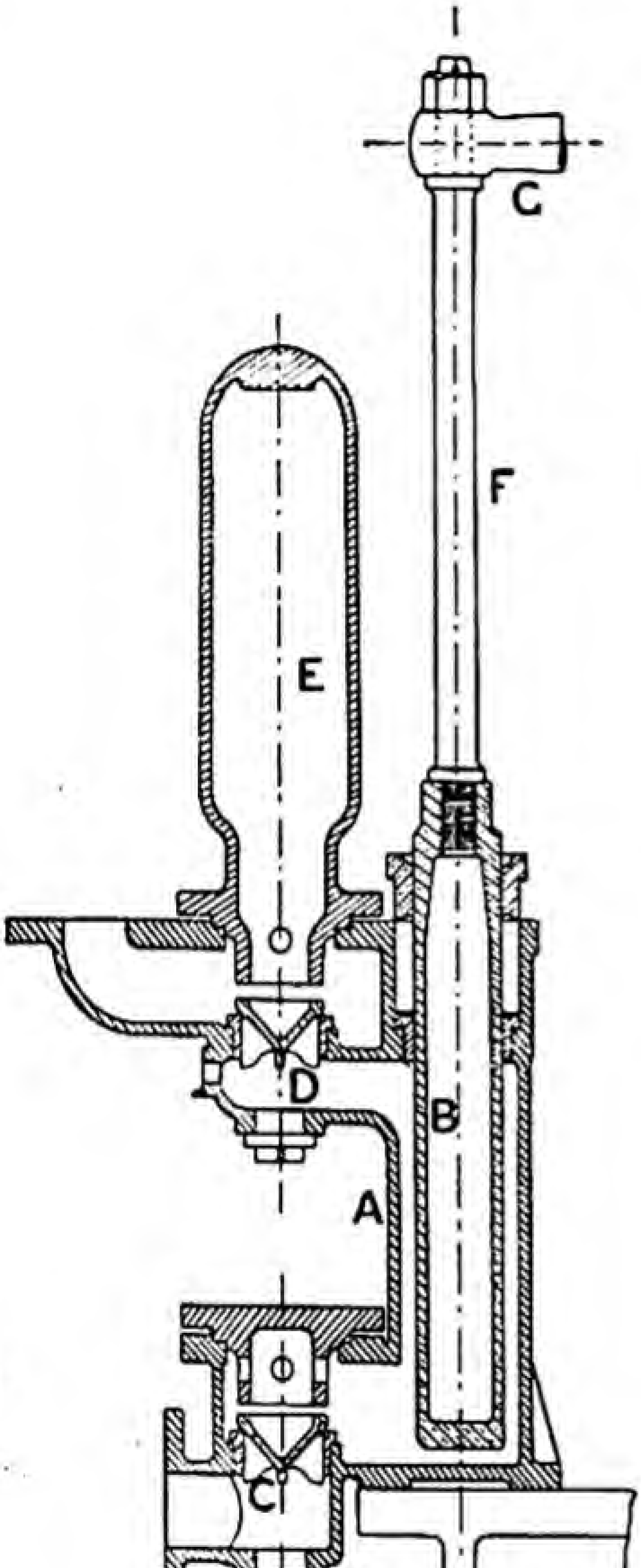


FIG. 231.—Marine feed pump.

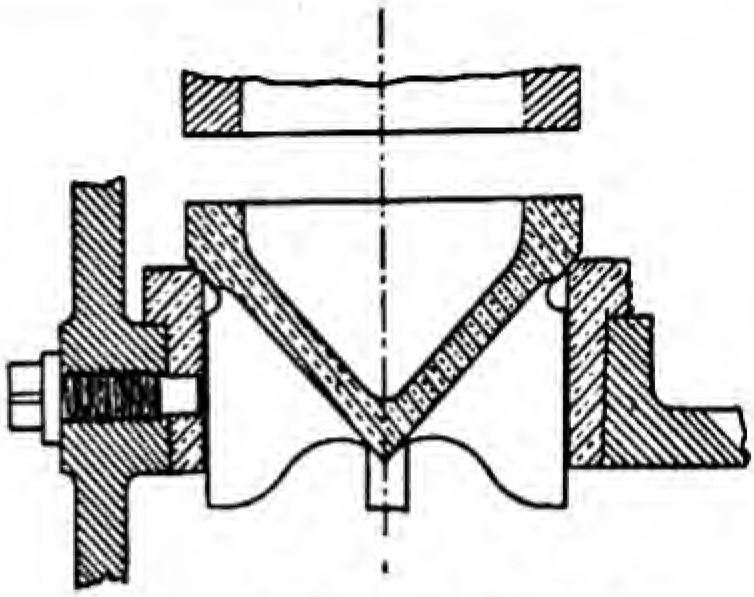


FIG. 232.—Marine feed pump valve.

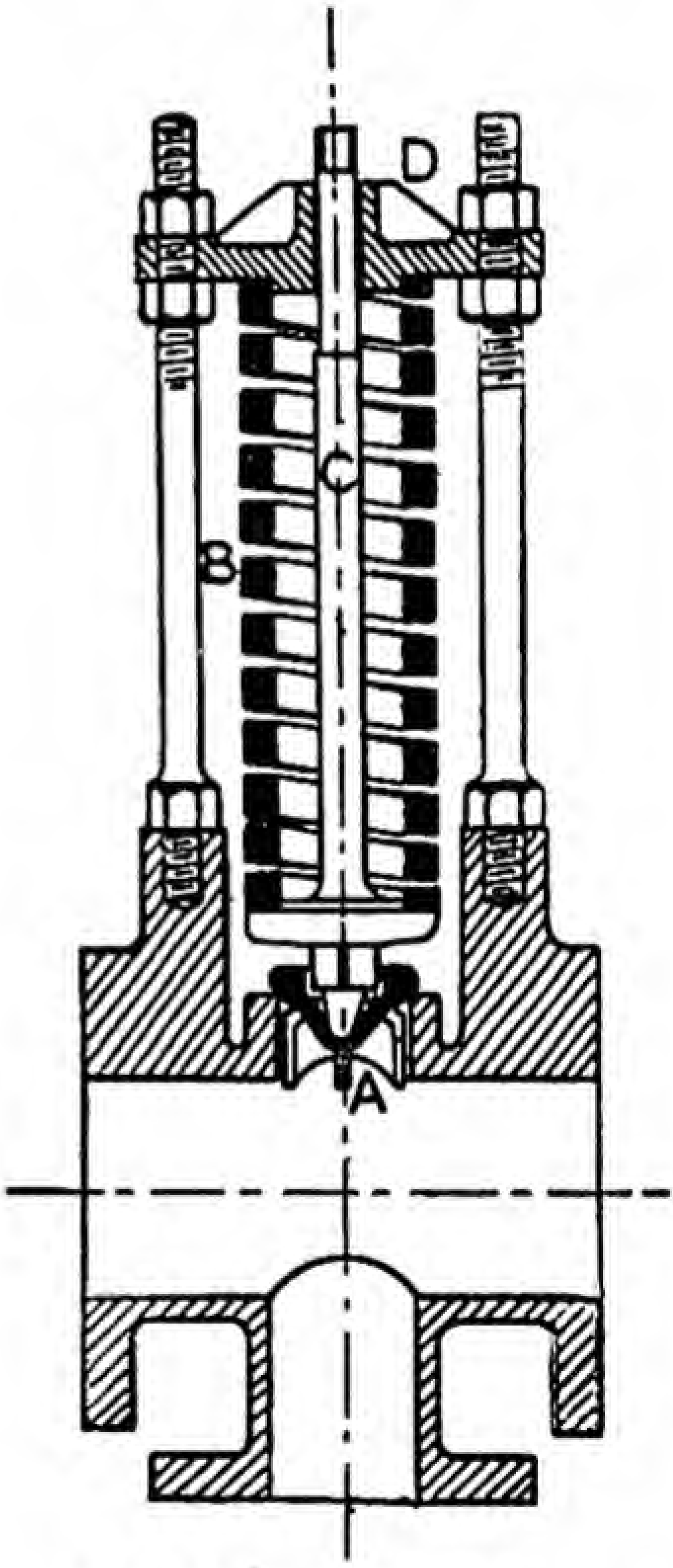


FIG. 233.—Marine feed relief valve.

a stuffing-box with brass gland and neck bush. The inlet valve is shown at C and the delivery valve at D. These valves are of the

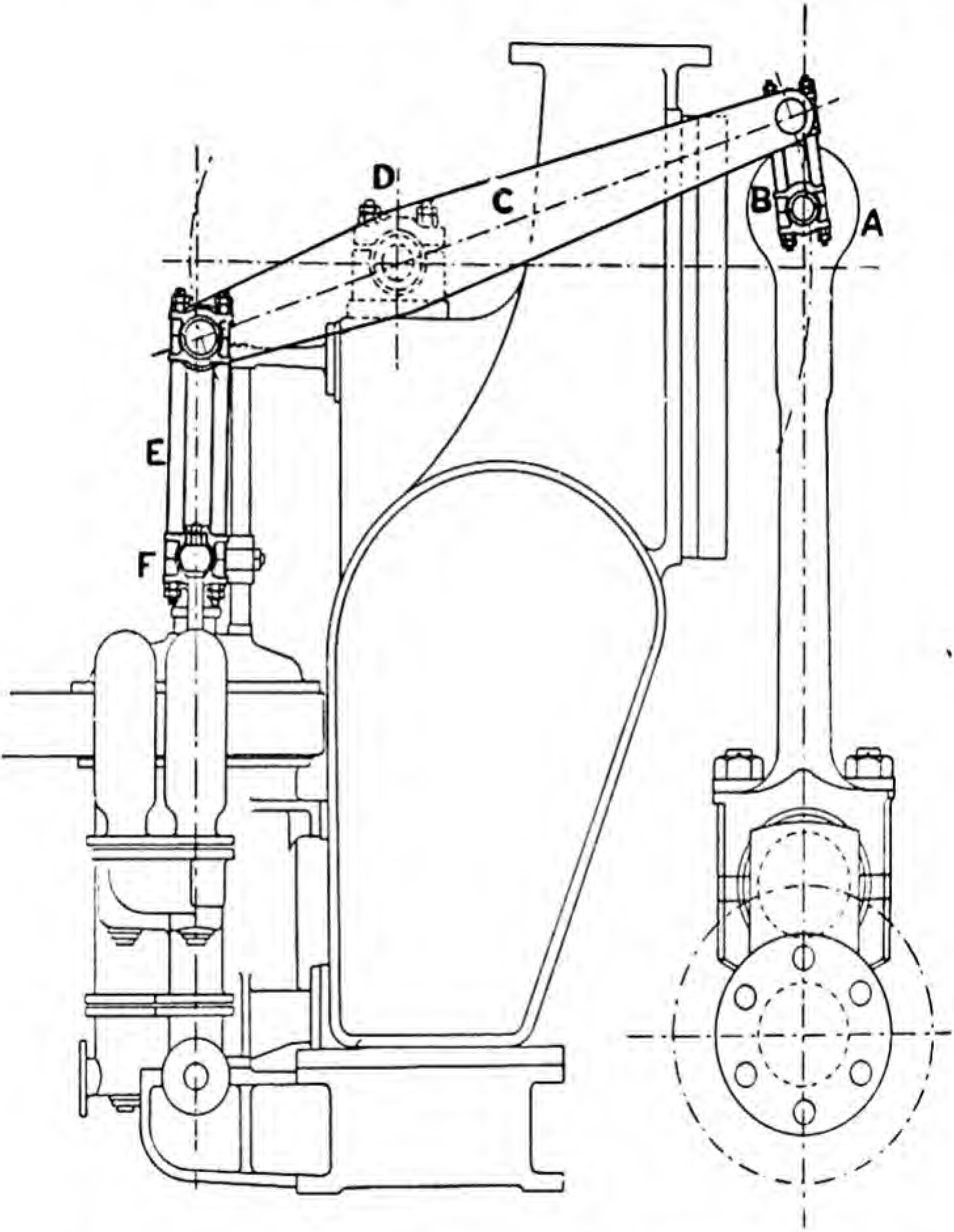


FIG. 234.—Arrangement for driving the air, feed and bilge pumps from the intermediate crosshead.

ordinary conical or clack variety and have a limited rise and fall. An **air vessel** is fitted at E. In working, during the up stroke the inlet valve C opens, permitting water to flow into the pump, the delivery valve being closed. During the down stroke, the inlet

valve closes and the water is forced by the plunger through the delivery valves D and so into the boiler. The air vessel contains some air, and this, by its elasticity, serves as a cushion for the reduction of shocks, which would otherwise be produced by the forceful pumping of incompressible water. The plunger is attached by means of a rod F to the air pump crosshead G, and so is driven by the main engines. A separate view of the pump valve and seat is shown in Fig. 232. The set screw for preventing the seat from becoming loose will be observed in this view.

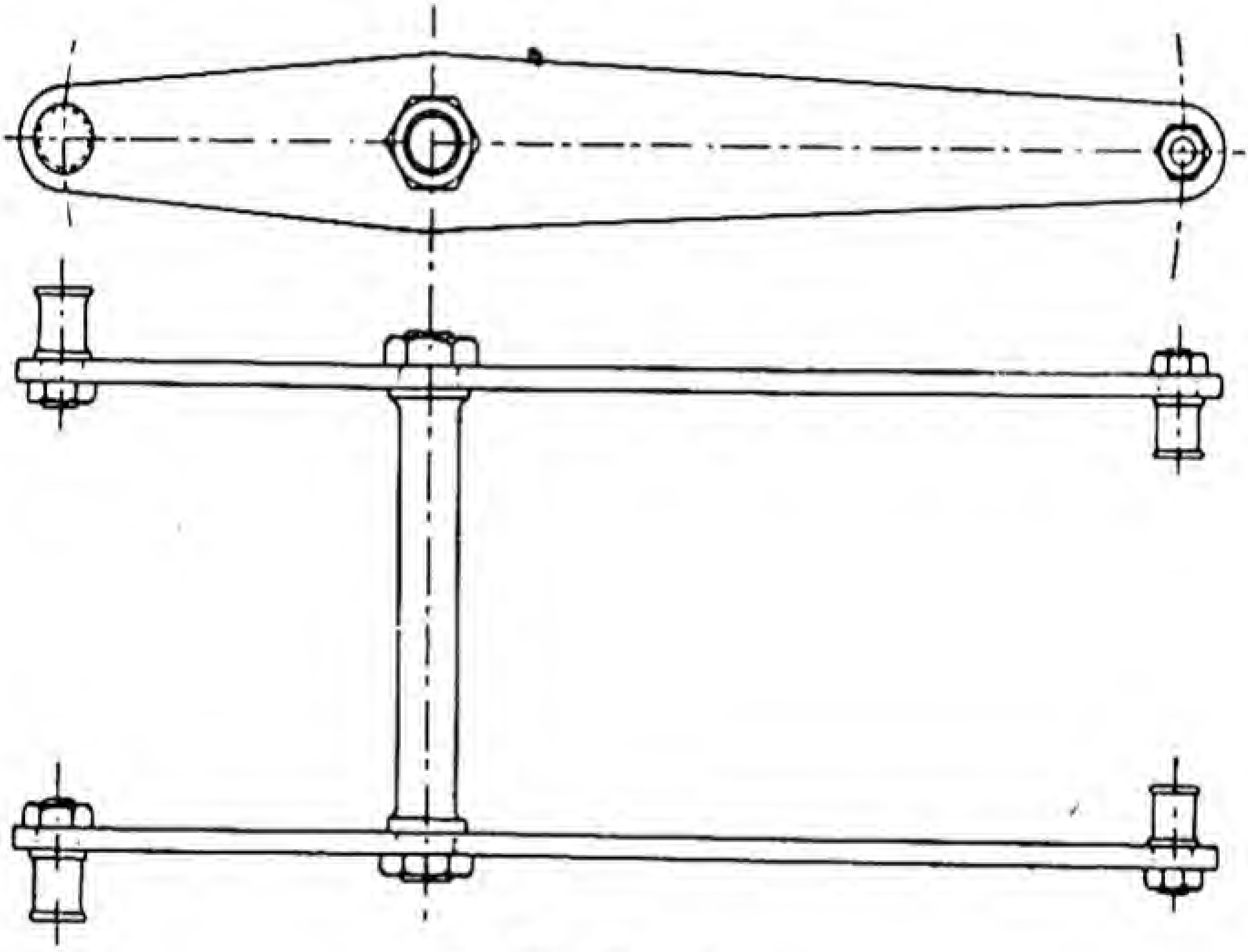


FIG. 235.—Pump levers.

Feed relief valve.—In the case of the engines still working and so driving the feed pump, damage might be caused by the shutting down of the boiler feed valves. To permit of the escape of the water discharged by the pump, a **feed relief valve** (Fig. 233) is fitted to the discharge pipe of the pump. This valve is similar in construction to the cylinder relief valve already described (p. 265). Provision is made for returning to the feed pump suction any water lost through the relief valve.

Arrangement of pump drive.—The method of driving the pumps is shown in Fig. 234. The intermediate crosshead pin

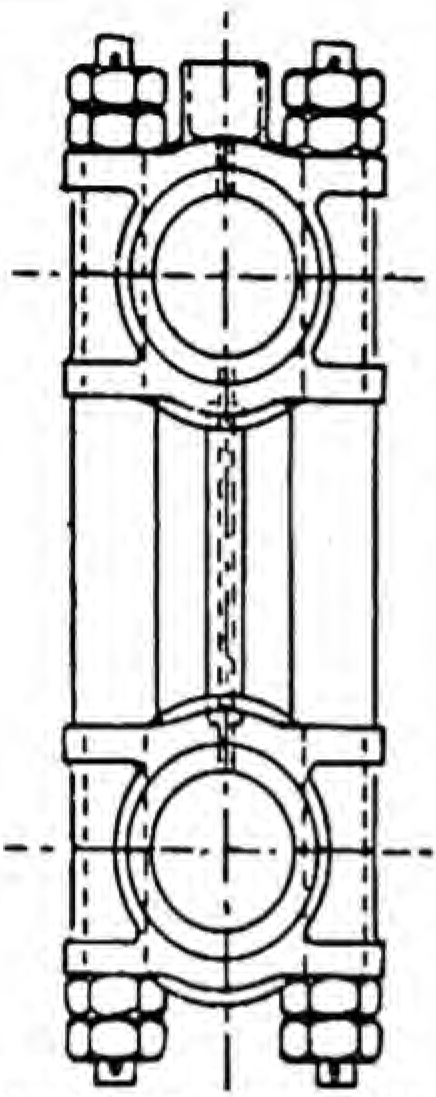


Fig. 236.—Pump connecting links.

A is connected by short **links** *B* to a **rocking lever** *C*, which has its bearings at *D*. The short end of the lever is connected by other **links** *E* to the pump crosshead *F*. The bearing on which the lever rocks is supported on a bracket cast on to the condenser and column. The air pump levers are made of steel plates (Fig. 235) with the necessary pins and gudgeon secured to them by nuts. The connecting links are shown separately in Fig. 236, and consist of gun-metal bearings connected by bolt stays. There are in all five pumps driven from the air pump crosshead, viz. the air pump, two feed pumps and two bilge pumps, used for clearing the bilges of water.

Circulating pumps.—The water for cooling the condenser is allowed to flow into the condenser from the sea, and is pumped overboard by means of

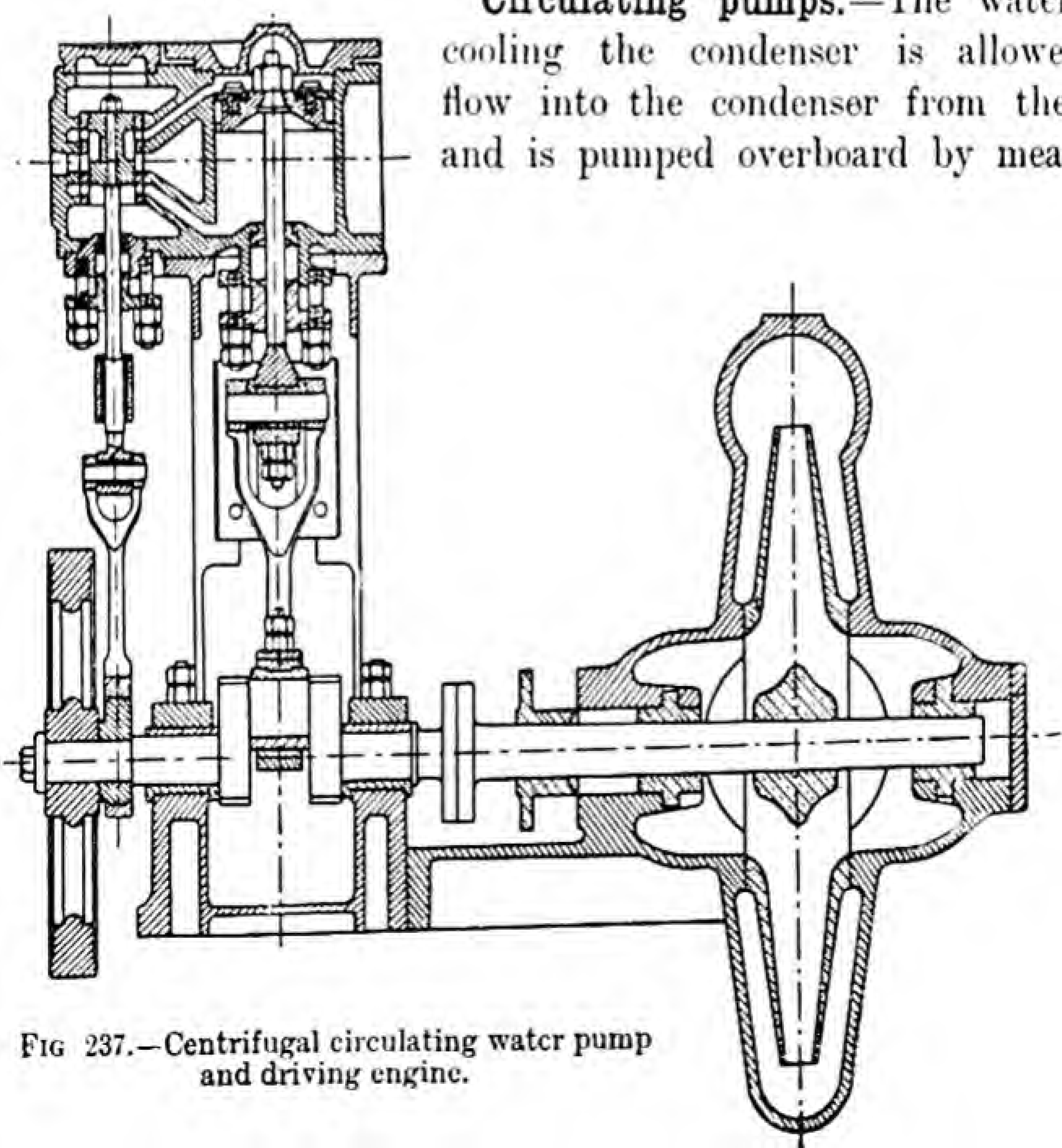


Fig 237.—Centrifugal circulating water pump and driving engine.

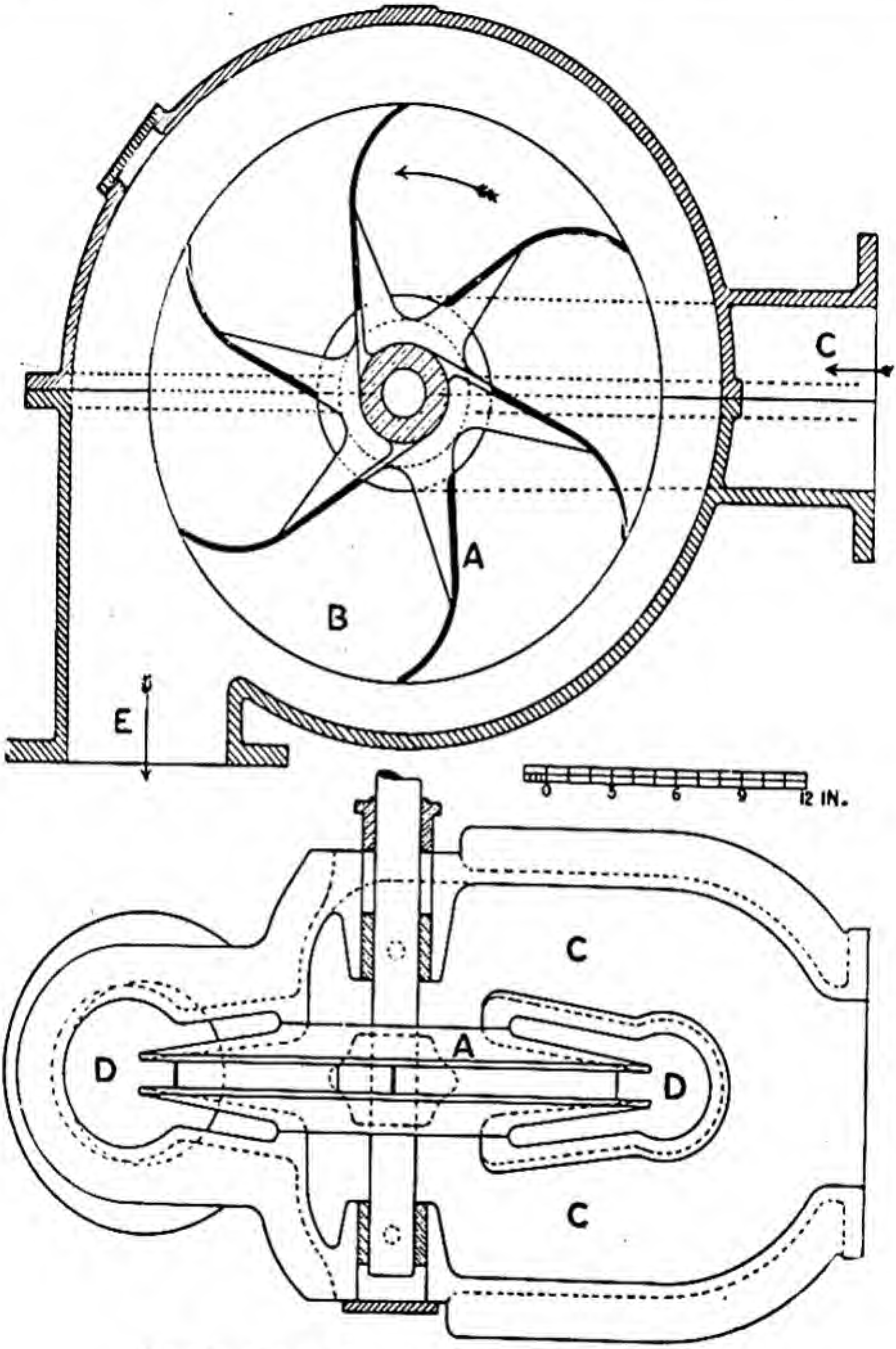


FIG. 238.—Details of centrifugal circulating water pump.

a **centrifugal pump** driven by an independent steam engine having a cylinder 5″ in diameter × 3¾″ stroke (Fig. 237). The pump is shown in detail in Fig. 238, where it will be observed that

it consists of a **six-bladed wheel** *A*, 24″ in diameter, rotating inside a chamber *B*. Water enters the chamber at the centre of the wheel through the inlet passages *C*, *C*, where the rotating wheel imparts additional kinetic energy to it, and so delivers it through the whirlpool chamber *D* to the discharge *E*. The driving side of the wheel shaft is provided with a stuffing-box; the other end is closed by a plate to prevent leakage. The wheel shaft is connected to the engine crank shaft by means of a flanged coupling (Fig. 237).

Frequently, not only the circulating pump but also the air pump is driven by an independent engine. The arrangement is advantageous in permitting a vacuum to be obtained in the condenser before starting the main engines, thus reducing the number of operations which have to be attended to at one time.

Quantity of circulating water.—Generally from 30 to 40 lbs. of circulating water are supplied to the condenser per pound of steam condensed. The quantity depends on the sea temperature, which of course is higher in tropical regions. With sea water at 60° F., and the water delivered to the hot well by the air pump at 120° F., about 15 lbs. of water condensed per square foot of cooling surface in the condenser may be considered fair practice.

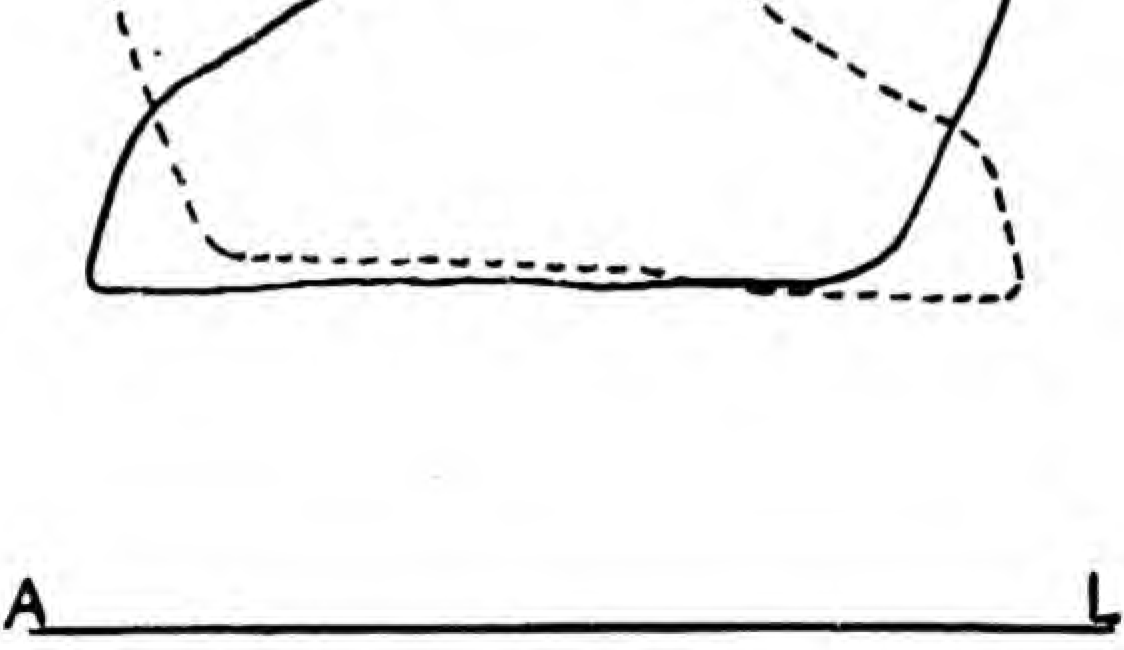


FIG. 239.—Indicator diagram from H.P. cylinder of triple expansion marine engines.

Indicator diagrams.—A set of indicator diagrams, obtained during a trial of the engines on 20th August, 1895, is given in Figs. 239,

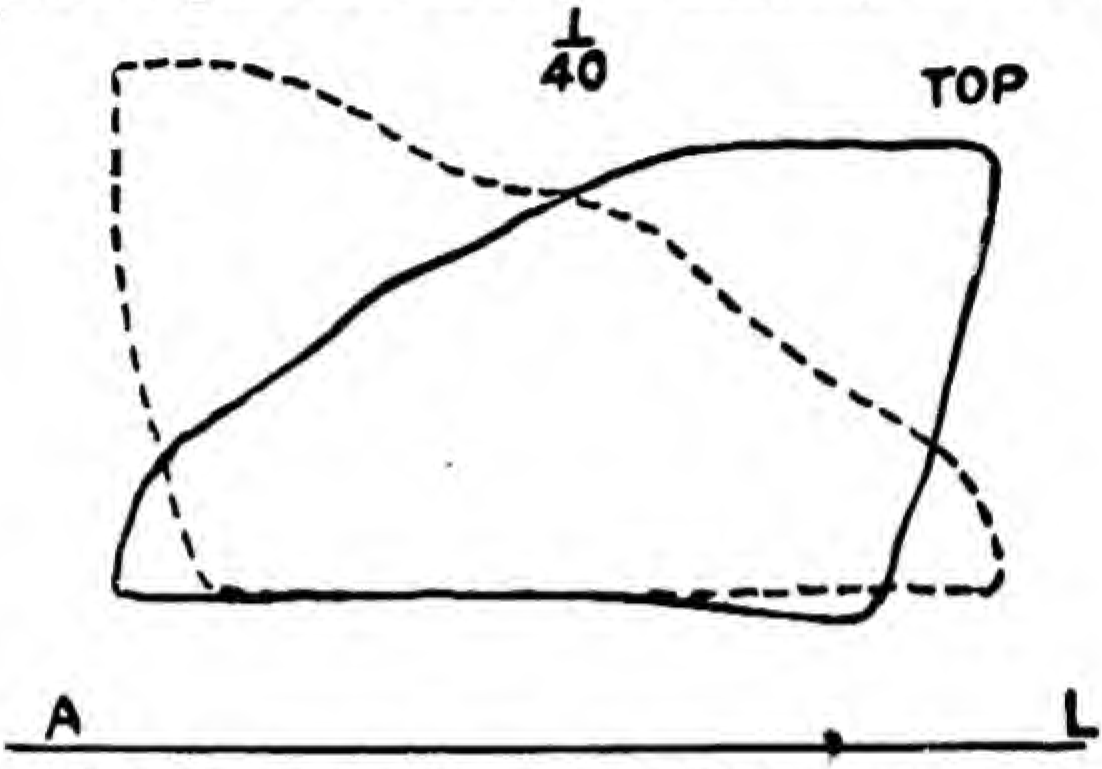


FIG. 240.—Indicator diagram from M.P. cylinder of triple expansion marine engines.

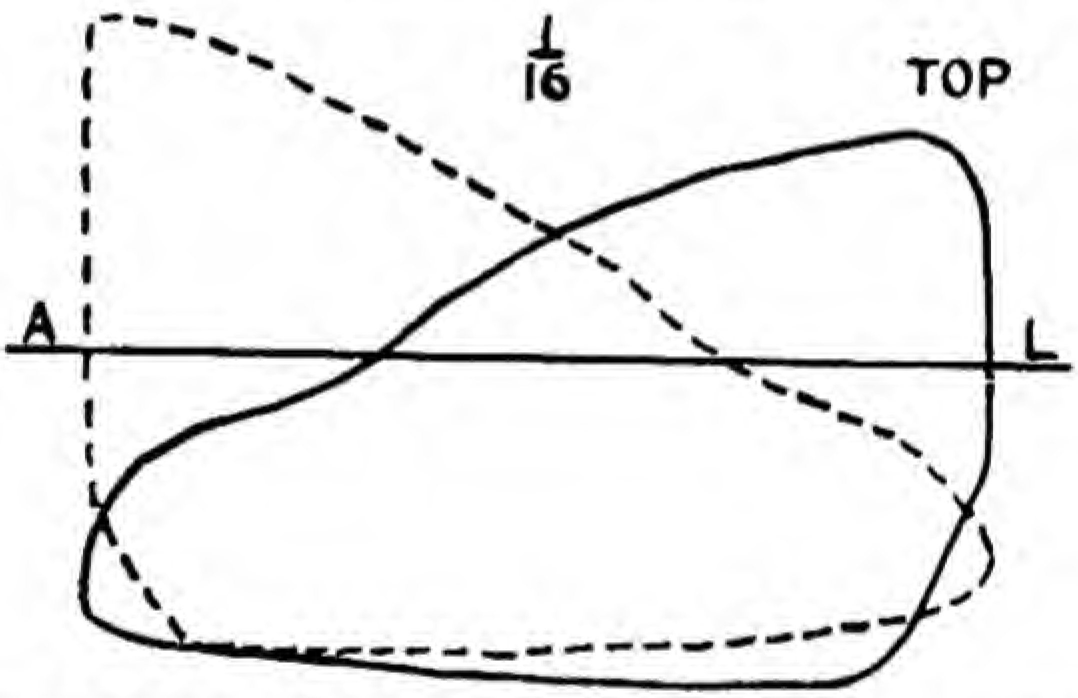


FIG. 241.—Indicator diagram from L.P. cylinder of triple expansion marine engines.

240, and 241. Other particulars obtained at the same time are as follows:

Steam pressure in boilers, - -	158 lbs. per sq. inch.
,, ,, ,, M.P. steam chest,	62 ,, ,, ,,
,, ,, ,, L.P. steam chest,	12 ,, ,, ,,
Vacuum in condenser, - - -	27¼ inches of mercury.
Revolutions per minute, - - -	143
I.H.P. from H.P. cylinder, - -	372·2
,, ,, M.P. ,, - - - -	460
,, ,, L.P. ,, - - - -	502
Total I.H.P., - - - - - -	1334·2

EXERCISES ON CHAPTER XV.

1. In the trial of the Bellis compound engine (p. 257) the following data were noted: diameters of cylinders, H.P. $=13''$, L.P. $=22''$, stroke $=11''$, revs. $=375$; for H.P. cylinder, I.H.P. $=167$; for L.P. cylinder, I.H.P. $=153$. Calculate what must have been the mean pressures for each cylinder, neglecting the effect of the piston rod areas.

2. Give sketches and description of any one of the slide valves of the triple expansion marine engine, p. 261.

3. Sketch and describe the construction of the H.P. marine engine cylinder, p. 263. Show clearly how the liner is fixed.

4. Taking the dimensions of travel of valve, steam lap and lead on top given in the Table, p. 262, for the H.P. cylinder, verify the point of cut-off given in the Table.

5. Answer question 4 for the bottom of the L.P. cylinder.

6. In the Table, p. 262, the total ratio of expansion is given as 8·45. Is this the actual ratio of expansion? If not, calculate the actual value.

7. Taking the quantities required from the Table, p. 262, calculate what force will be required to overcome the inertia of the H.P. reciprocating parts, at the top and bottom of the stroke, when running at 145 revolutions per minute.

8. Sketch and explain the construction of a marine engine piston.

9. Give sketches and describe a marine crosshead and guides.

10. Sketch and describe the construction of the big end of a marine connecting rod.

11. Explain and give sketches of the method of moving, at the same time, all the links of the reversing gear.

12. Sketch and describe in detail a double bar link and block for a reversing gear.

13. What is the function of the thrust block in a marine engine? Sketch one of the shoes for such a block, and show how it is adjusted.

14. Give sketches and explain the construction of a propeller.

15. Explain why it is of importance that sea water should be kept out of the steam space of a marine surface condenser. What would be the effect of such leakage?

16. Describe with detailed sketches the connection of the tube plate to the condenser body, and also the method of packing the tubes.

17. Give sketches and describe the construction of an air pump bucket and valves.

18. Take sea water to weigh 64 lbs. per cubic foot, and calculate how much air per day would enter a marine boiler with the make up water, supposing 1500 lbs. of such water to be used per day.

19. Sketch and describe a feed pump intended to be driven by the main engines. What occurs to the delivery water, supposing the boiler feed check valves are shut down?

20. Show by sketches the method adopted in marine engines for driving the pumps from one of the main engine crossheads.

21. Give sketches and describe the action of a centrifugal circulating water pump.

22. Sketch and describe the escape valve as fitted to the cylinders of a marine engine. What is the use of such a valve? Show, by a sketch, where it is fixed. 1896.

23. Describe and sketch the construction of a double-beat or equilibrium valve. When and for what purpose are such valves used? In such a valve the two seats measure respectively 8 inches and $7\frac{1}{2}$ inches in diameter, and the weight of the valve is 70 lbs. What pressure per square inch would cause the valve to lift, the pressure between the valve discs being disregarded. 1896.

24. Sketch and describe the construction of the air pump bucket with its valves and packing, and show how it is worked in connection with a jet condenser. Of what materials are the body of the bucket and of the valves respectively made? 1897.

25. Explain and show, with sketches, the construction and action of the force pump employed for feeding the water into a boiler when an injector is not used. Sketch also in section the "clack" or non-return valve attached to the boiler. How is the pump prevented from forcing water into the boiler when the engine is running, but a supply of water is not required? 1897.

26. The ram of such a pump (question 25) is 2 inches in diameter, and has a stroke of 24 inches. How many gallons of water (neglecting leakages) would be forced into the boiler for each 1000 double strokes (one forward and one backward) of the pump?

1 gallon of water = 0·16 cubic feet. 1897.

27. Describe, with sketches, a large air-pump for a steam engine. 1907.

28. Steam enters a cylinder at any initial (absolute) pressure p_1, it is cut off at $\frac{2}{5}$ of the stroke. What is the average pressure during the stroke? It is some fraction of p_1. Assume the hypothetical diagram, no clearance, an expansion law pv constant. Apply your answer to the cases where p_1 is 100, 80, and 60. If the back pressure is 17, what is the mean effective pressure in each case?

The area of the piston is 300 sq. inches, crank 2 feet; two strokes in a revolution; what is the work done in one revolution in each of the above cases? Tabulate your answers. 1907.

29. Two strokes in a revolution; area of piston 300 sq. inches; crank 2 feet. What is the volume (neglecting clearance) of steam admitted if the cut-off is at $\frac{3}{8}$th of the stroke? If the initial pressure is 100 or 80 or 60 pounds per square inch, what weight of steam is used in one stroke (assuming no condensation, no clearance). What weight is used in one revolution?

p	100	80	60
Volume in cubic feet of 1 lb. of steam,	4·356	5·37	7·03

1907.

CHAPTER XVI.

LOCOMOTIVES.

Locomotives.—There are two principal types of **locomotives**, one intended for running on rails, the other intended for traction purposes on ordinary roads. Dealing with the first type of locomotives, these consist in general of an engine having a pair of cylinders, the pistons of which are connected by the usual crank and connecting rod mechanism to a crank shaft having a pair of cranks placed at right angles. The crank shaft has a pair of driving wheels, one at each end, which are properly constructed so as to run on the rails. The engines are mounted on a frame, on which also is mounted a boiler, situated over the engines. The whole is supported on wheels, springs being placed between the axles and the frame in order to reduce shocks. Frequently there are four, six, eight or even ten driving wheels, these being mounted on axles in pairs, one axle being driven direct by the engines and the others being connected to the first by means of coupling rods and cranks placed at right angles. The locomotive is said to be four, six, etc. coupled, depending on the number of driving wheels.

Although compound locomotives are used largely, the more general plan is to have both cylinders high pressure. The exhaust steam is delivered up the chimney and is so utilised for securing the necessary draught of air through the furnace. **Inside cylinder** locomotives have the cylinders placed between the frames, thus necessitating the use of cranked driving axles. **Outside cylinder** locomotives have the cylinders placed outside the frames, working on crank pins secured to the driving wheels, the driving axles being straight in this arrangement. The tractive effort of the

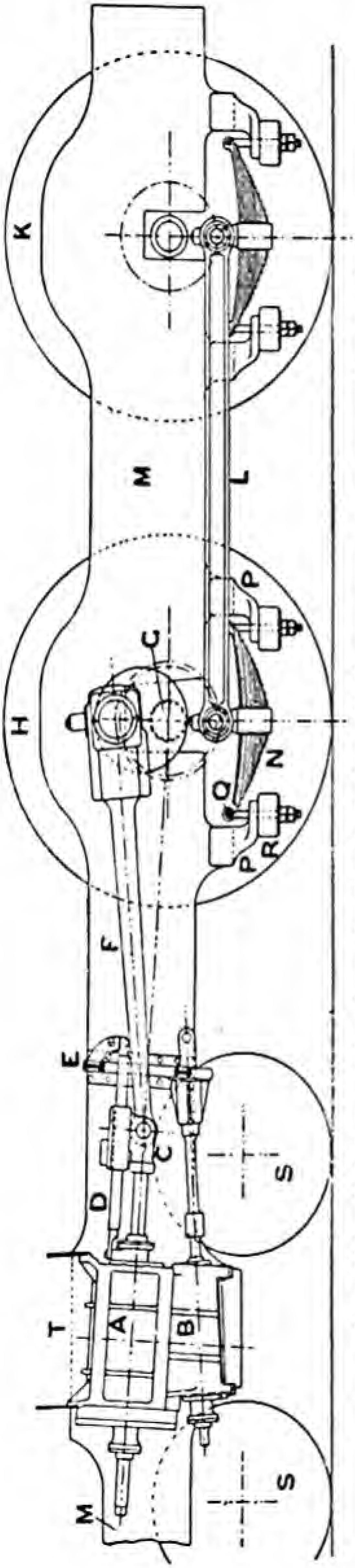


FIG. 242.—Driving mechanism of a Great Eastern Railway express passenger locomotive.

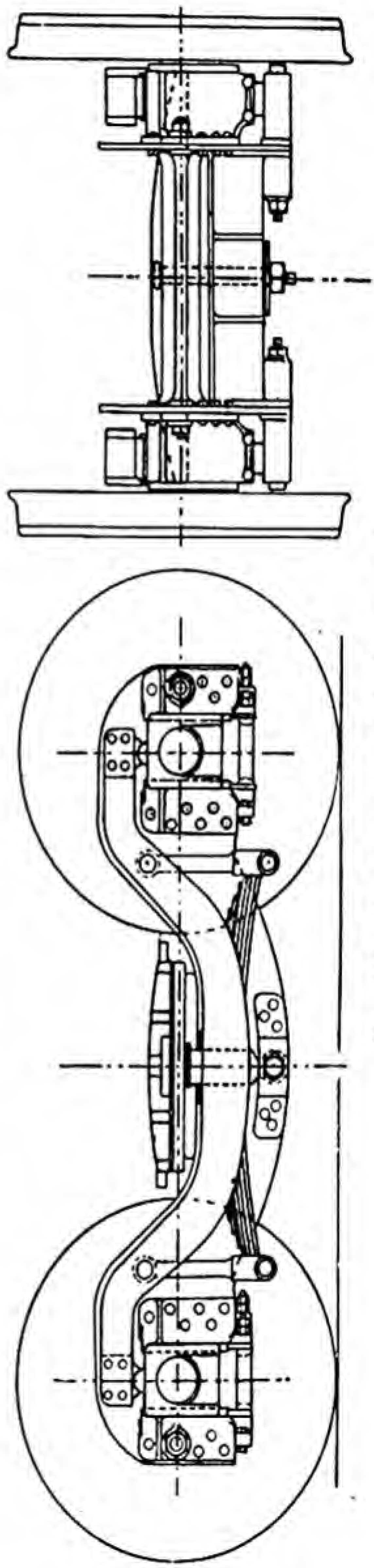


FIG. 243.—Arrangement of the bogie truck.

locomotive is secured by reason of the friction between the driving wheels and the rails. There must therefore be a sufficient proportion of the weight of the locomotive carried by the driving wheels in order to secure the required frictional adhesion.

Engines having a great length of wheel base generally have the front part of the locomotive carried on a **bogie** in order to give flexibility in rounding curves and taking points. The bogie is simply a small car with four wheels, connected to the frame by a central vertical pin, on which the bogie can swivel and so adjust itself to the rails.

The valves are usually of the simple slide valve type, having the backs fitted with some form of relief frame. Generally in this country Stephenson's link motion is the gear used.

By the courtesy of Mr. James Holden, some of the details of a Great Eastern Railway express passenger locomotive are given here. The locomotive is four coupled, of the inside cylinder type. The boiler has already been described in Chapter X.

The engine mechanism.—The arrangement of the principal parts of the mechanism will be understood by reference to Fig. 242, which shows the connections to one of the cylinders, the valve gear being omitted. The cylinder *A* has the valve chest *B* placed underneath and is securely bolted to the side frames *M*. The piston rod is connected to a crosshead *C*, which slides on a single guide bar *D*, the latter being bolted to the cylinder at one end and to the motion plate *E* at the other end. The motion plate consists of a casting running between the two side frames *M* and is bolted to them at each of its ends; it serves to carry one end of the guide bars and also guide brackets for the valve spindles. The connecting rod is shown at *F* and the driving axle at *G*. The driving wheels *H* and *K* on the side of the locomotive shown are coupled by the coupling rod *L*. A similar coupling rod connects the pair of wheels on the other side of the locomotive.

The driving axles run in axle boxes, which rest on coach springs *N*; these springs are slung from brackets *P*, bolted to the side frames, by means of suspending rods *Q*. The rods pass through cases *R*, containing several rubber washers, which are put under compression by the weight of the machine and assist in damping vibrations. The weight of the front part of the locomotive is taken by a bogie having four wheels, two of which are indicated

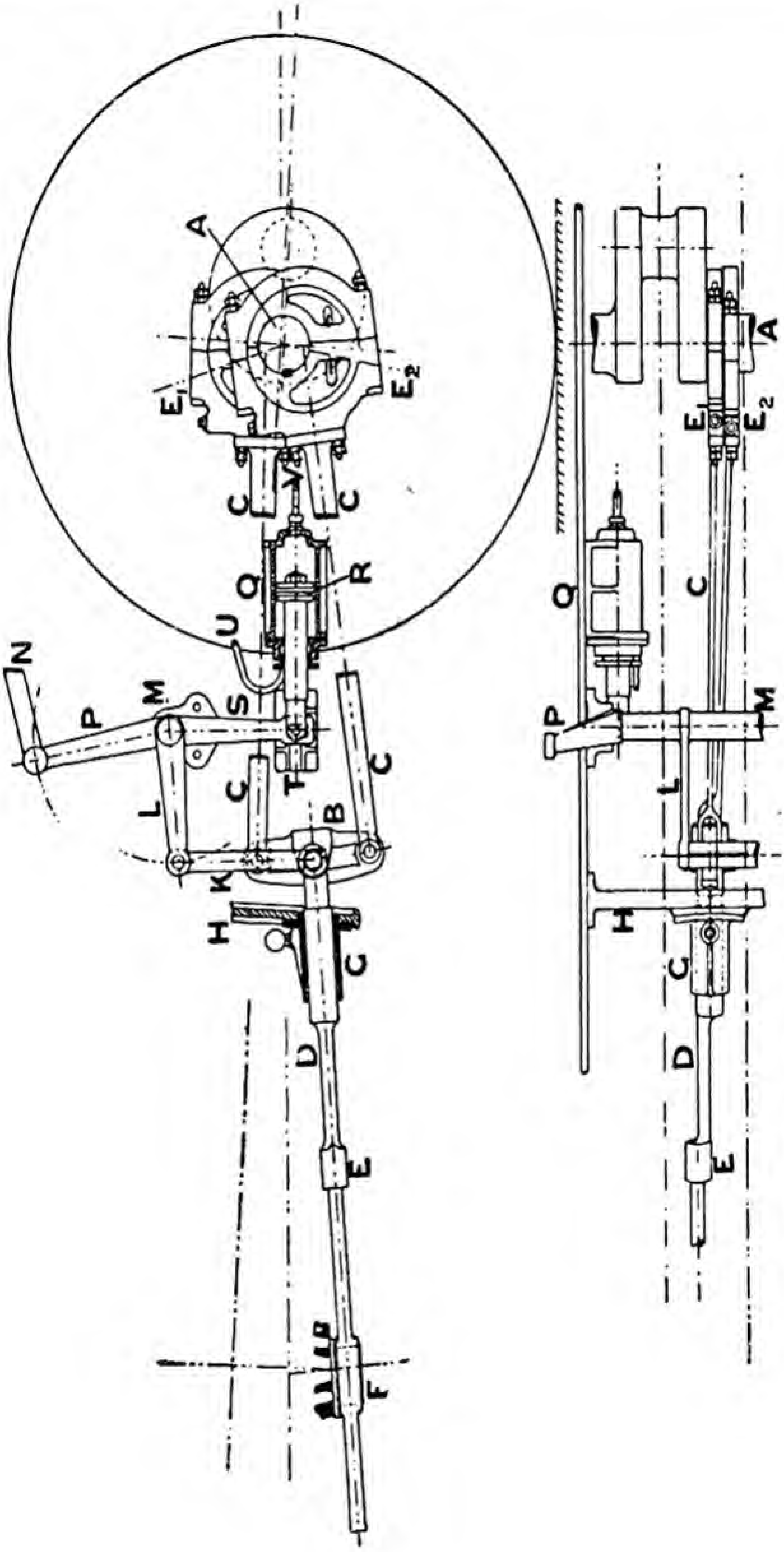


FIG. 244.—Valve gear of a Great Eastern Railway express passenger locomotive.

at *S*, *S*. Details of the bogie truck and its springs are shown in Fig. 243.

The valve gear.—The valve gear is of the Stephenson link motion type, the arrangement being illustrated in Fig. 244. The eccentrics E_1 and E_2 are secured to the driving axle *A* and are connected to the link *B* by means of eccentric rods *C*, *C*, which are shown broken in the illustration. The details of the eccentrics are illustrated separately in Fig. 245. The valve rod *D* is in two parts,

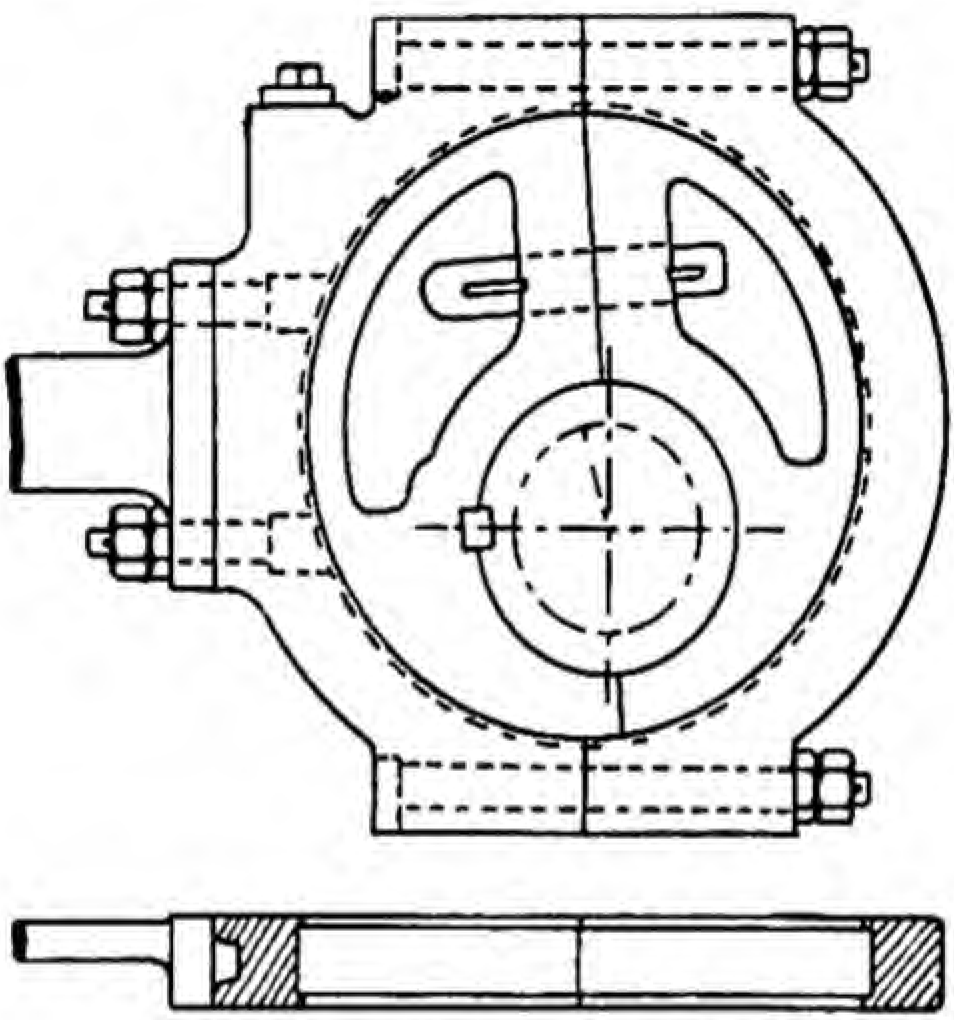


FIG. 245.—Locomotive eccentric.

connected at *E* by a cotter joint, and is connected to the valve at *F*. The valve rod works in a guide *G* secured to the motion plate *H*, and carries at its outer end a block which slides in the slotted link *B*. The link is suspended by lifting rods *K* from a lever *L* on the reversing shaft *M*. The link of the other motion is connected in a similar manner to a lever fixed to the same shaft. Both links are put over simultaneously by operating the reversing shaft through the rod *N*, connected to the reversing lever at the foot plate and to the lever *P* on the reversing shaft.

An air cylinder is fixed to the side frame at *Q*, its function being to support the weight of the links and half the weight of the eccentric rods, and also to supply a means of reversing the engine

by power, thus reducing the manual effort which must otherwise be applied to the reversing lever. The piston *R* is attached to a piston rod of large diameter, which is connected at its outer end to the lever *S* fixed to the reversing shaft. The piston rod is guided by a block sliding in a bracket *T* secured to the side frame. Pipes for supplying air under pressure are connected to each end of the cylinder at *U* and *V*, and lead to a valve not shown. In the running position both ends of the air cylinder are in communication with the main air reservoir, the effect being to produce a resultant force on the piston urging it towards the left. The ratio of the diameter of the piston rod to that of the cylinder is adjusted so that the resultant force may be sufficient to support the weight of the motion hanging from

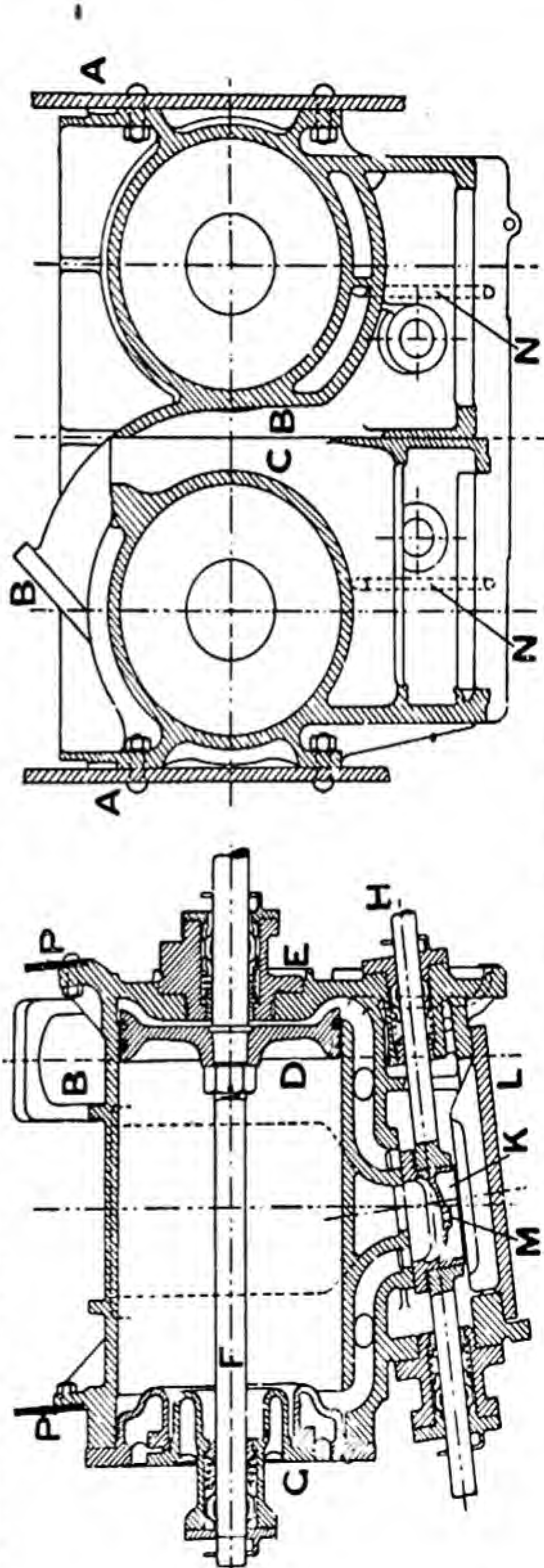


FIG. 246.—Sections of Great Eastern Railway locomotive cylinders.

the links K. In reversing, the valve is operated so as to put one side of the cylinder in communication with the exhaust, the other end being still supplied with air under pressure, thus causing the piston to move from one end of the cylinder to the other and so to raise or lower the links.

The cylinders.—The cylinders are cast together in one piece with the steam chests underneath (Figs. 246 and 247). The cylinders are strongly bolted to the side frames at A, A. Steam is led into the steam chests through the passage B and is exhausted to the blast nozzle through the passage C. The piston D is of pressed steel secured to the piston rod—which is tapered to receive it—by means of a nut with locking pin. The piston rod passes through the cover E to the crosshead and is furnished with a tail rod F passing through the cover G. The valve rod H has also a tail rod, and is connected to the slide valve K by means of a buckle forged on the rod. The slide valve is balanced, being furnished with a packing ring sliding on a planed surface of the steam chest cover L, and provided with a hole M which puts the space at the back of the valve into communication with the exhaust port, thus relieving the back of the valve from the pressure of the steam. The cylinder ports are shown in plan in Fig. 248.

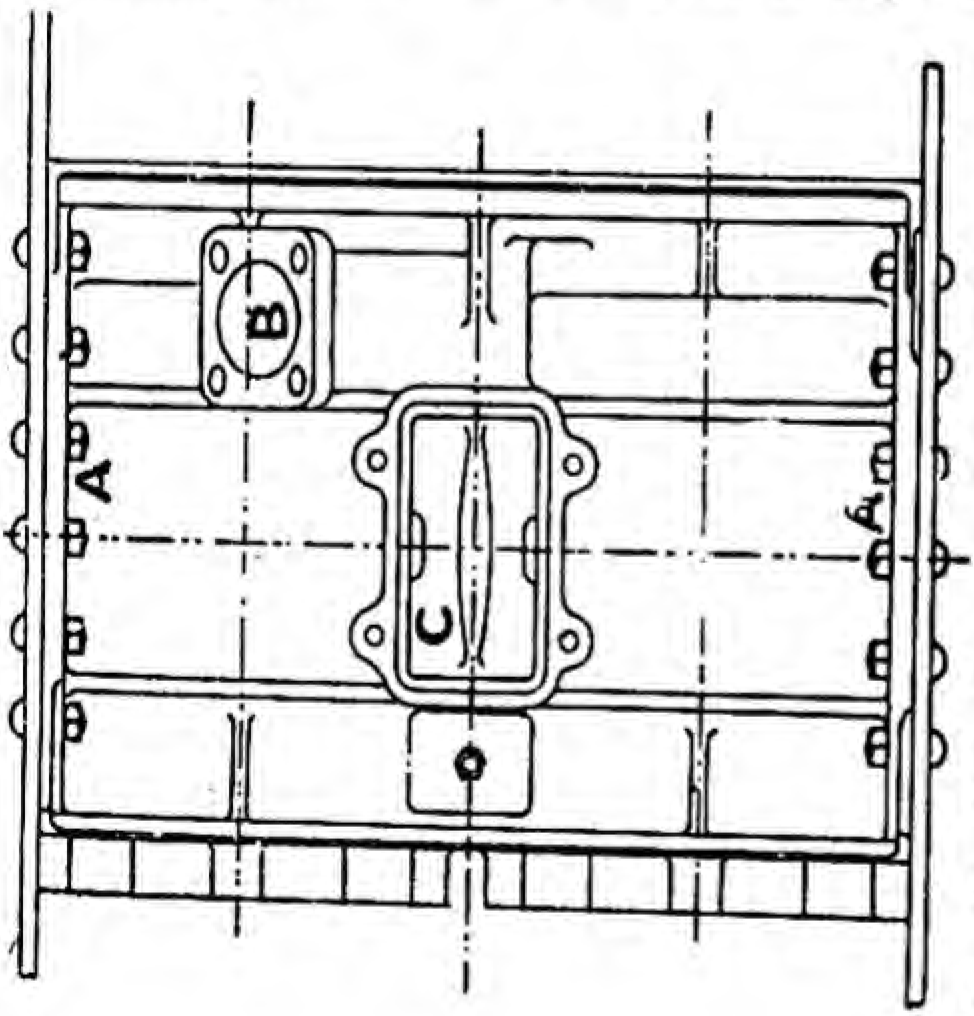


FIG. 247.—Plan of Great Eastern Railway locomotive cylinders.

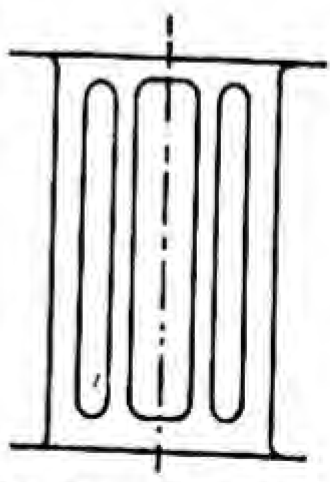


FIG. 248.—Plan of the cylinder ports.

The piston is packed with two Ramsbottom spring rings fitted into grooves in its rim. The piston rod and valve rod are rendered steam-tight with metallic packing fitted to the stuffing-boxes. Drainage of water from the cylinders is provided for by means of passages *N*, *N* leading from the steam ports to the lowest part of the cylinders, where the drain cocks are situated. The smoke box is secured to the cylinders at *P*, *P*.

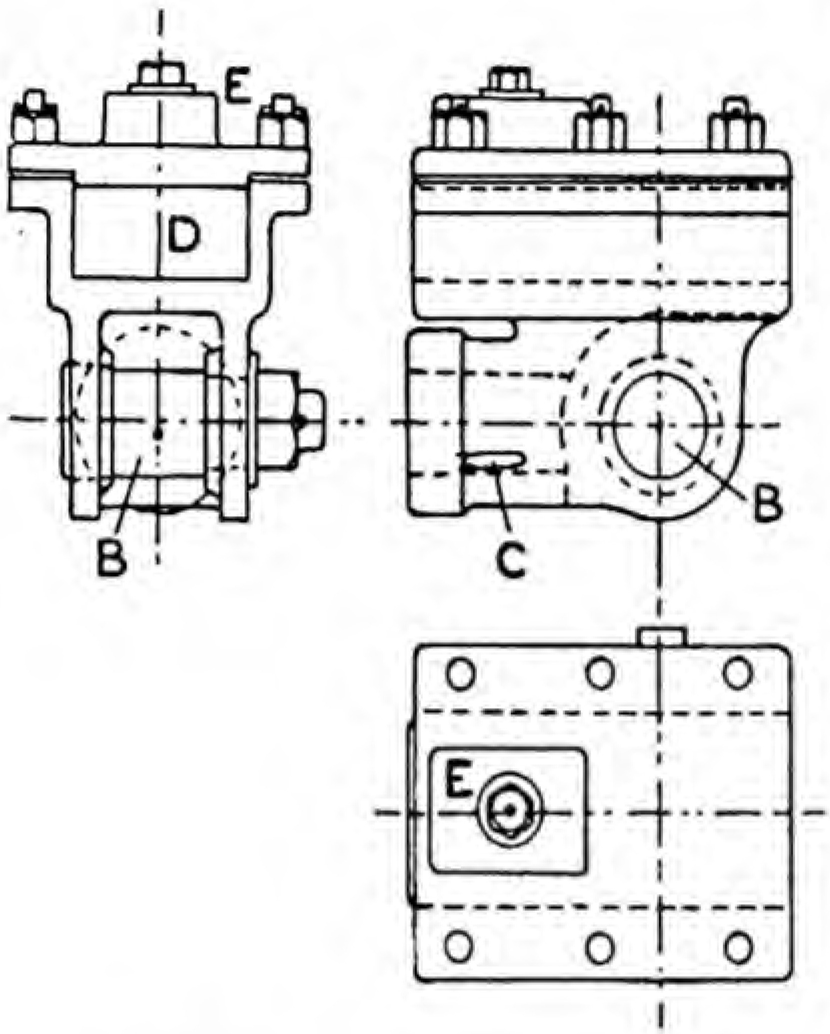


FIG. 249.—Locomotive crosshead.

Crosshead and connecting rod.—The crosshead consists of a casting (Fig. 249) cored out to receive the connecting rod, the joint being made by the gudgeon pin *B*. The crosshead is secured to the piston rod by a cotter at *C*, the piston rod end being fitted to the tapered hole in the crosshead. The crosshead is formed to embrace completely the slide bar at

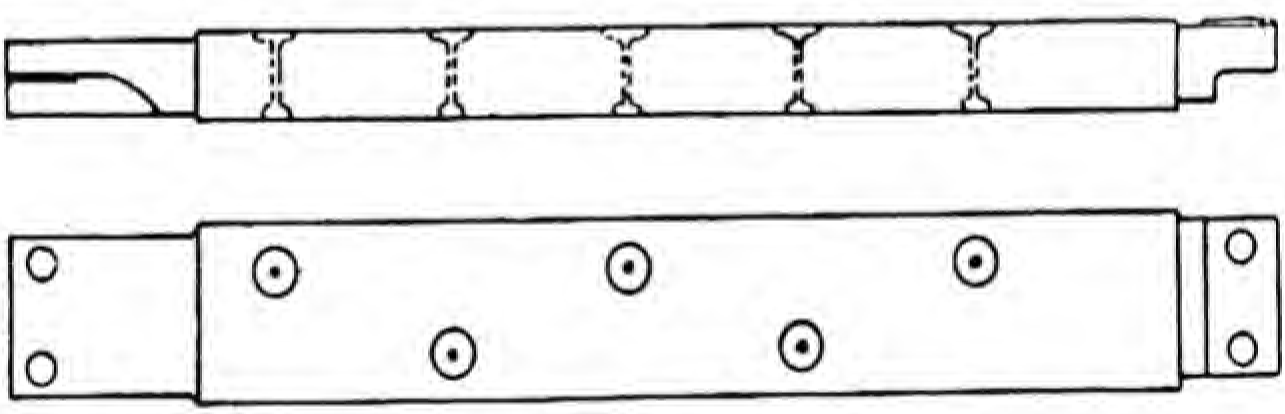


FIG. 250.—Locomotive slide bar.

D. In this type a single slide bar is fitted, in other types double slide bars are often employed. The slide bar is shown separately in Fig. 250. Oil is supplied to the slide bar from the oil box *E* on the top cover of the crosshead and finds its way to the under side of the crosshead through the oil holes shown in Fig. 250.

The connecting rod is illustrated in Fig. 251. It is of rectangular

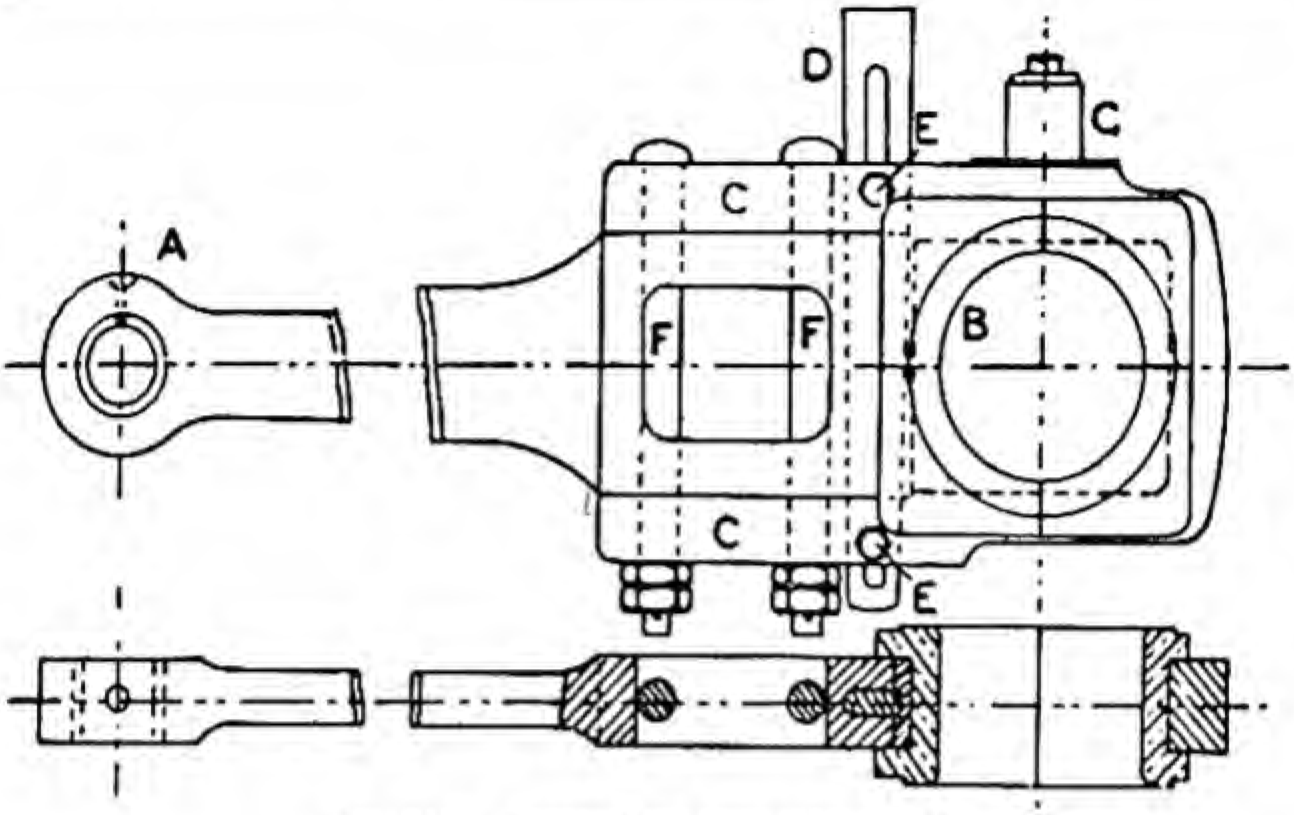


FIG. 251.—Locomotive connecting rod.

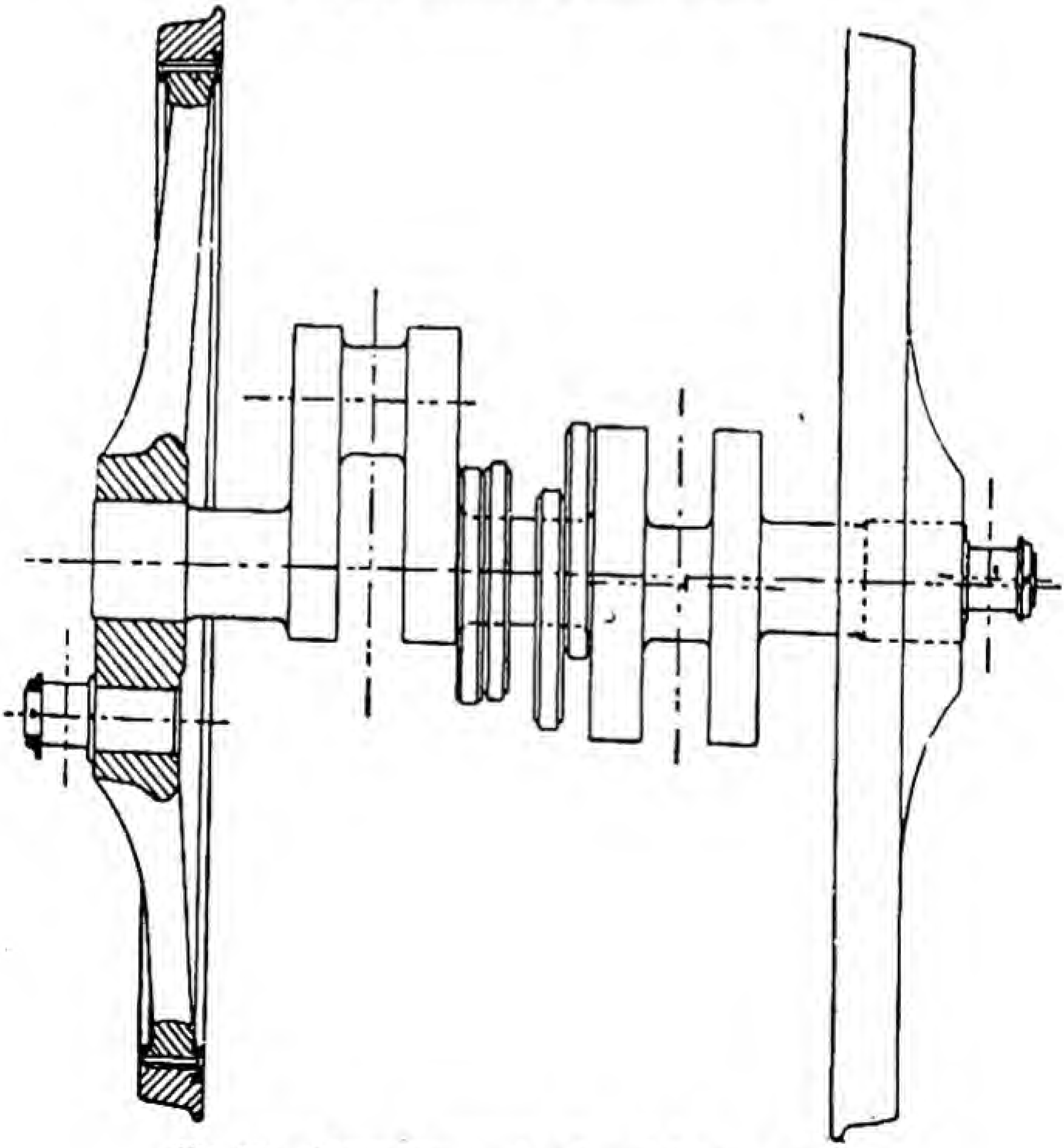


FIG. 252.—Locomotive crank axle and driving wheels.

cross section, tapering from the crank pin end to the gudgeon end. The gudgeon end is fitted with a plain gun-metal bearing *A*; the crank pin bearing *B* is of gun-metal and is in halves, held in position by a strap *C*. Wear is taken up by a cotter *D*, which is locked by two set screws *E*; the strap is secured to the rod by means of two bolts *F*, *F*, with lock nuts and split pins. Oil is supplied from the oil cup *G*.

Crank axles.—These are shown in Fig. 252, together with a pair of driving wheels. The axle is made out of the solid with circular crank webs. The cranks are placed at 90°. The eccentric sheaves are keyed on between the cranks and therefore have to be made in halves, as is shown in Fig. 245, the halves being secured together by a pin with two cotters. The driving wheels are of cast steel with steel tyres secured to the wheel by the method illustrated in Fig. 253. The tyre has a projection *A* on the outside of the wheel; a ring *B* is placed on the inner side and the whole is held in position by rivets. The object is that the tyre, should it break, may be still held in position on the wheel. Tyres are turned to a slightly conical form after they are in position on the wheels. When running round a curve, the flange of the wheel on the outer rail of the curve comes against the rail, the wheel thus running on its largest diameter, while the wheel on the inner rail is running on its smallest diameter. The object is to reduce the slip, and consequent wear to the wheels, which would otherwise result by reason of the outer wheel having a greater distance to travel in rounding the curve than the inner, while both wheels make the same number of revolutions.

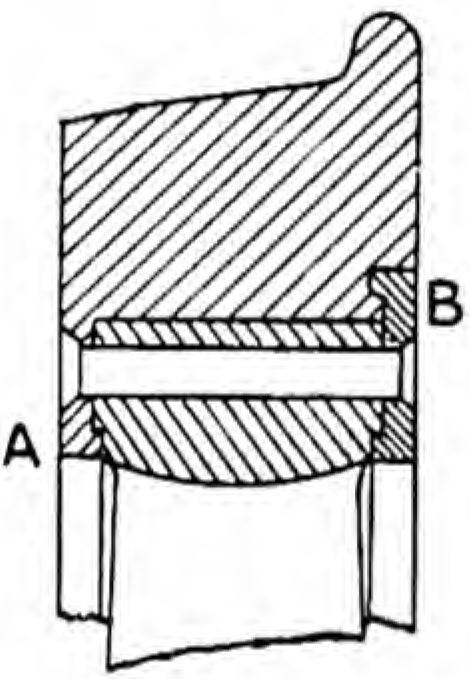


FIG. 253.—Method of securing locomotive tyres.

Crank pins for the connection of the coupling rods are shown secured to the wheels in Fig. 252. These are forced into holes bored to receive them, and then riveted over.

Balancing.—When running, the mechanism of the engine sets up inertia forces, which must be reduced so far as possible by balancing. This is accomplished by casting large balance weights

with the driving wheels, as shown in Fig. 254. Want of balance in the horizontal direction produces impulsive effects on the draw bar. In the vertical direction, want of balance will produce, during each rotation of the driving wheels, an effort tending to lift the wheel from the rail at one place, thus causing slip and wear at the

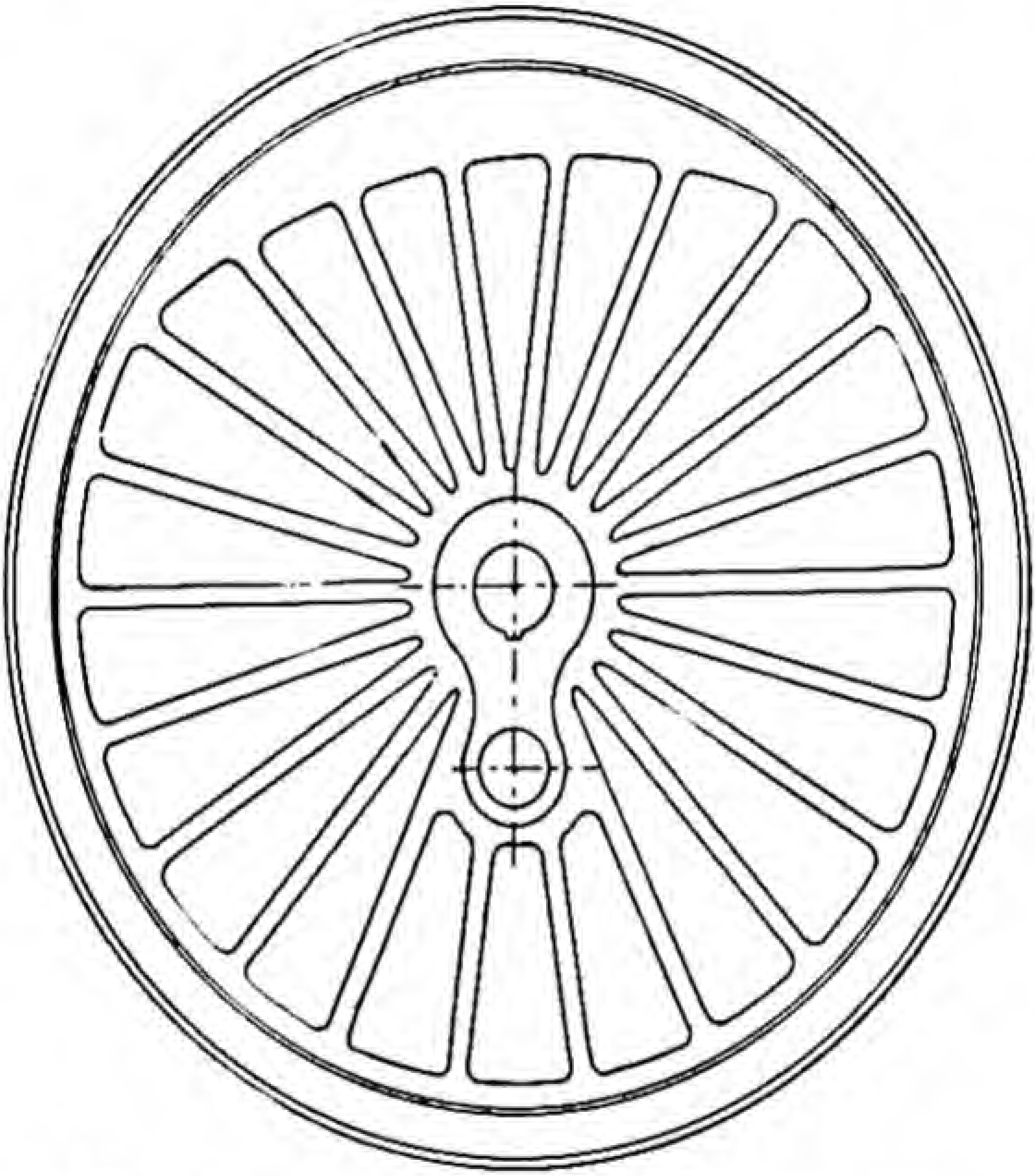


FIG. 254.—Locomotive driving wheel and balance weight.

part of the tyre which is then in contact with the rail; at another place there will be an impulsive downward effort on the wheel, producing a hammer blow on the rail.

Liquid fuel in locomotives.—The Great Eastern Railway locomotives are fitted with apparatus for the consumption of oil fuel. Mr. James Holden has taken a prominent part in the development of this method of raising steam. In Fig. 255, *A* is one of two injectors for spraying the mixture of coal tar and green oil, oil gas tar, creosote or other oil used as fuel into the furnace. These injectors are inserted in orifices in the front of the boiler,

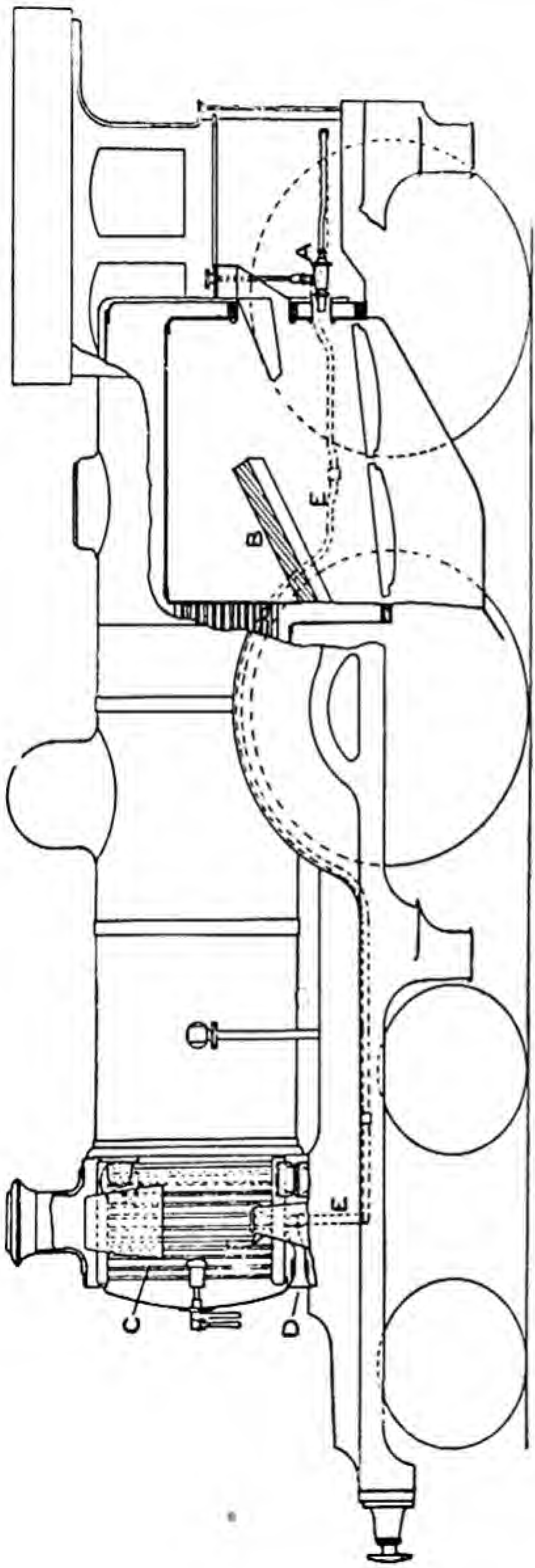


FIG. 255.—Oil fuel arrangement on a Great Eastern Railway express passenger locomotive.

one on each side, and are so disposed that they do not interfere with the use of coal in the furnace. The construction of the Holden injector is shown in Fig. 256. The object is thoroughly to break up the oil fuel into a spray of fine particles. There are three internal cones in the injector, having finely adjusted spaces between. The oil is fed into the outer space, and on emerging from the end of the cone at *A* is met by a mingled jet of steam and air coming from the other spaces. Air is supplied through the centre and steam between the inner and middle cones. On emerging from the mouth of the injector at *B*, the mingled jet of air, steam and partially atomised oil, is met by jets of steam coming from a ring *C*. These jets effectually complete the atomising, with the result of securing complete combustion of the fuel.

The steam required to run the injector is taken from the boiler. The jet action of the steam flowing through the nozzles enables it to draw in the air supply through the central orifice; indeed the vacuum, which may be maintained by the injector, has been applied for the working of the vacuum brakes. Regulation of the oil supply is accomplished by means of valves operated by hand wheels under the control of the driver.

Coal is used for raising steam in the boiler. The furnace is fitted with a specially large fire-brick bridge *B* (Fig. 255), towards which

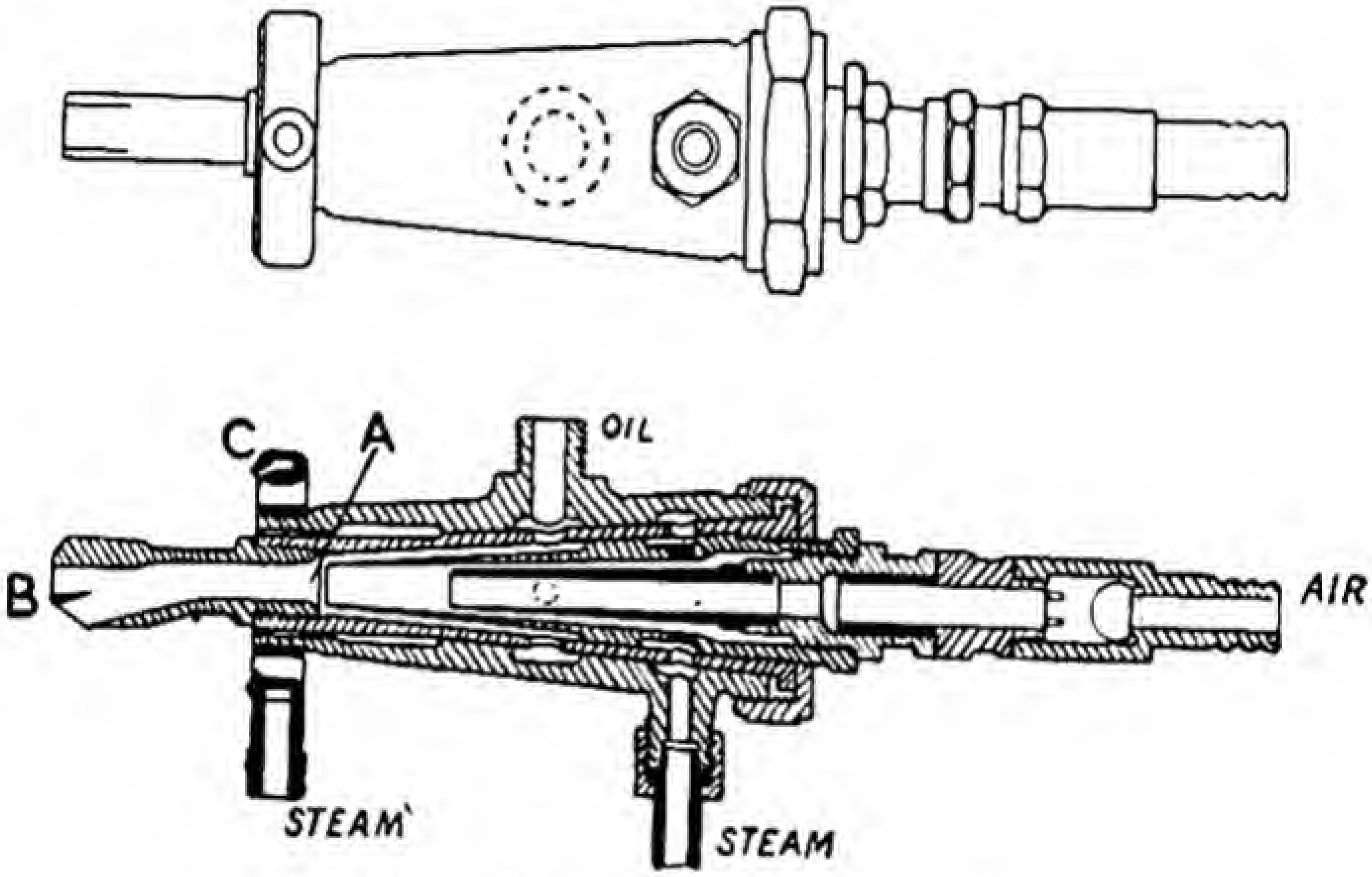


FIG. 256.—Holden's liquid fuel injector.

the injector jets are directed. This bridge intercepts and vaporises any oil which has escaped being atomised by the injectors. A thin layer of incandescent fuel is kept on the fire-bars during running. The air required for the injectors may be supplied hot by drawing it through a steam heating arrangement consisting of a number of pipes *C* placed in the smoke box. The air enters through the orifice *D*, and after passing through the heater is led through the pipe *E* to the injectors. The oil supply is contained in tanks placed in the tender and connected by pipes to the injector.

The arrangement renders it possible to use coal firing alone or oil firing alone, or both coal and oil may be burned simultaneously. The alteration from one system to the other can be effected, while

the engine is actually running. The adaptability of the Holden system has led to its adoption all over the world.

Oil fuel has also been applied to marine boilers, and there are many vessels now employing this fuel.

Road locomotives.—There are many varieties of locomotives now in use on ordinary roads for conveying goods. Many of these are driven by oil motors, others by steam, the boilers being fired in some cases by coal, in others by oil fuel. While the earliest locomotives in this country ran on highways, the development of a light machine capable of running at a fair speed dates from the removal of legislative restrictions in 1898. At the present date sufficient time has not elapsed for anything like a standard type to be produced excepting in traction engines which are of great weight and of slow speed. Road locomotives of the latter class have been in use for a long period. Generally a boiler of the locomotive type is used, with the engine placed over the boiler and geared to the driving wheels.

Steam waggon.—The following description is of a steam waggon designed by Mr. T. W. Barber, to whom, and to the Editor of "Engineering" the author is indebted for the illustrations. The waggon is designed to comply with colonial requirements, and is capable of undertaking severe duty. The general arrangement is shown in Fig. 257, where *A* is the steam boiler, placed in front of the leading axle with the condensers *B* behind it. The engines are situated at *C*, and consist of two independent sets of engines, each set driving one of the rear wheels by means of a chain. The exhaust steam from the engines passes through an oil separator *D* on its way to the condenser.

There are two systems of brakes, one consisting of rim brakes *G* on each rear wheel, the other consisting of expanding brakes attached to the hubs. These are operated respectively by the hand wheel *H* and the foot lever *J*. Clutches permit the engines to run disconnected from the rear wheels. The control system consists of three hand levers, one for compounding the engines, one for reversing and one for regulating the steam supply.

The boiler. The boiler is illustrated in sectional elevation in Figs. 258 and 259. There is a steam chamber *d* and two water headers *e* connected by a number of brass tubes expanded and ferruled into the tube plates. The tubes are bent as shown in

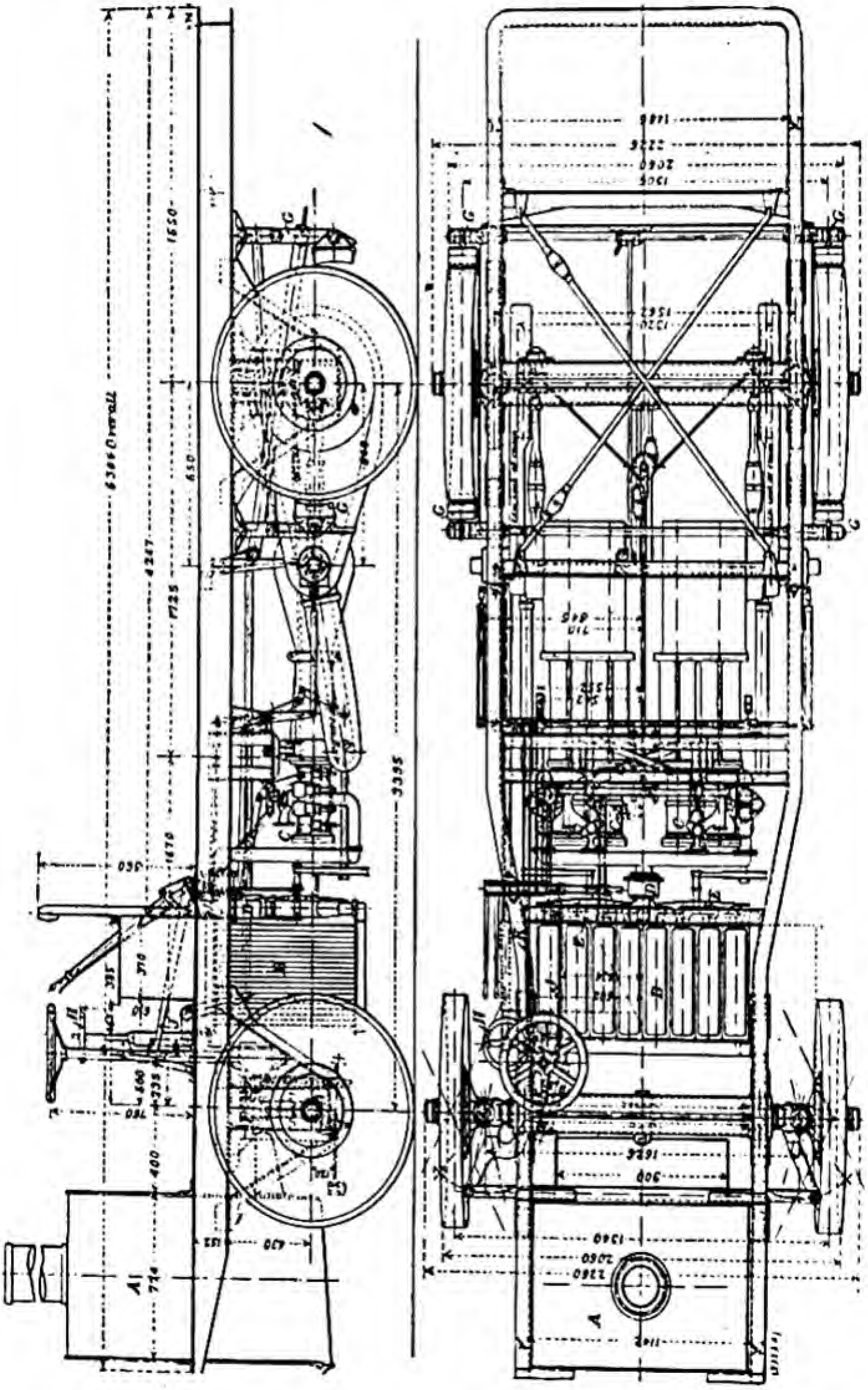


FIG. 257.—General arrangement of Barber's steam waggon.

Fig. 262 to obtain maximum efficiency of heating surface. The steam chamber and headers are steel castings strengthened by

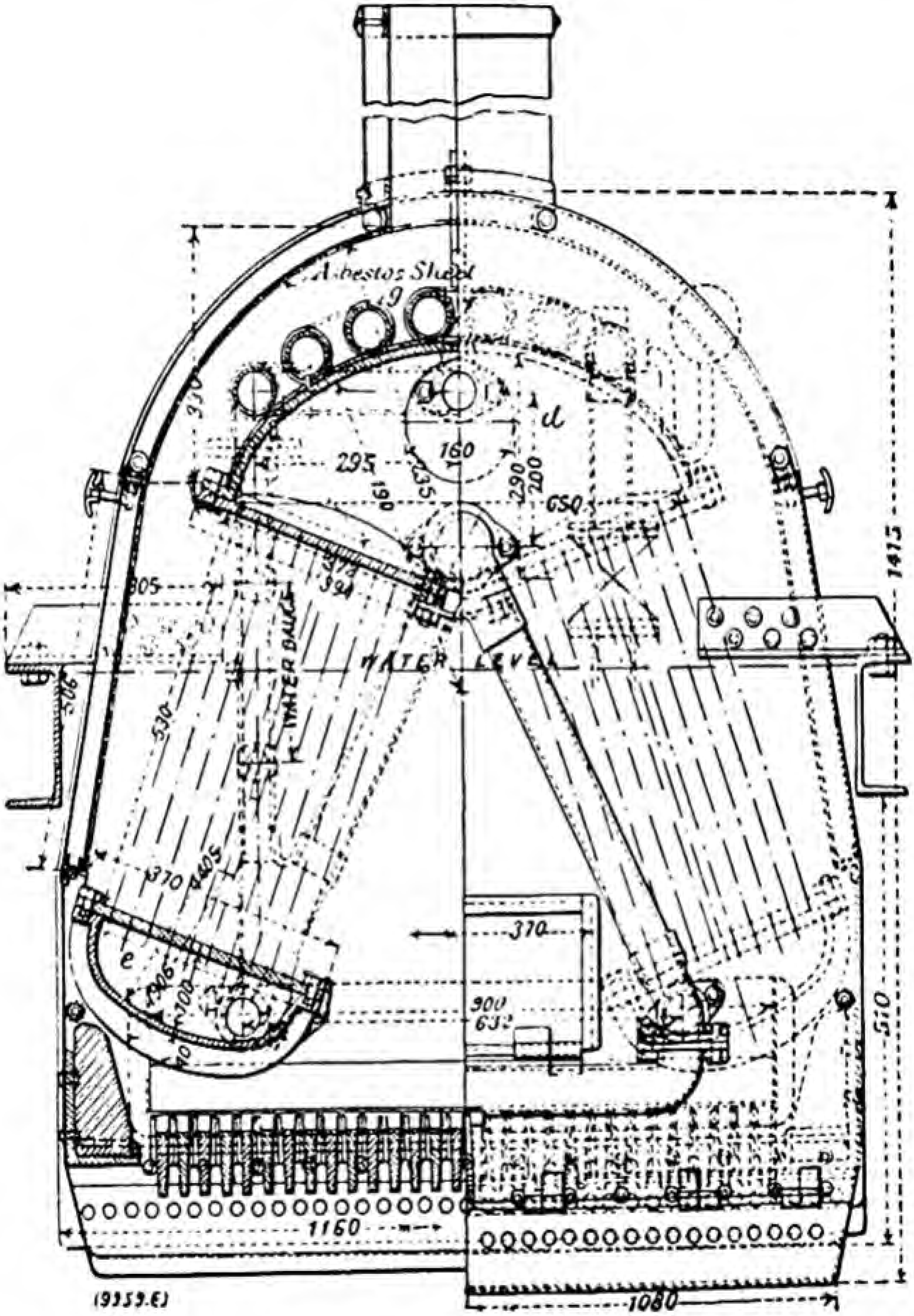


Fig. 258.—Sectional front elevation of the water-tube boiler for Barber's steam waggon.

ribs: the internal ribs of the steam chamber have each a circular hole to permit free passage of steam. There are two downcomer tubes h one at each end of the boiler, to promote circulation of the water. The fire-box accommodation is large, to permit the use of

wood and other indifferent fuel being used for steam raising without forcing. The steam when formed is collected by a perforated pipe and passes out of the steam chamber into a pipe *f* leading into a series of tubes *g*. These tubes are heated by the

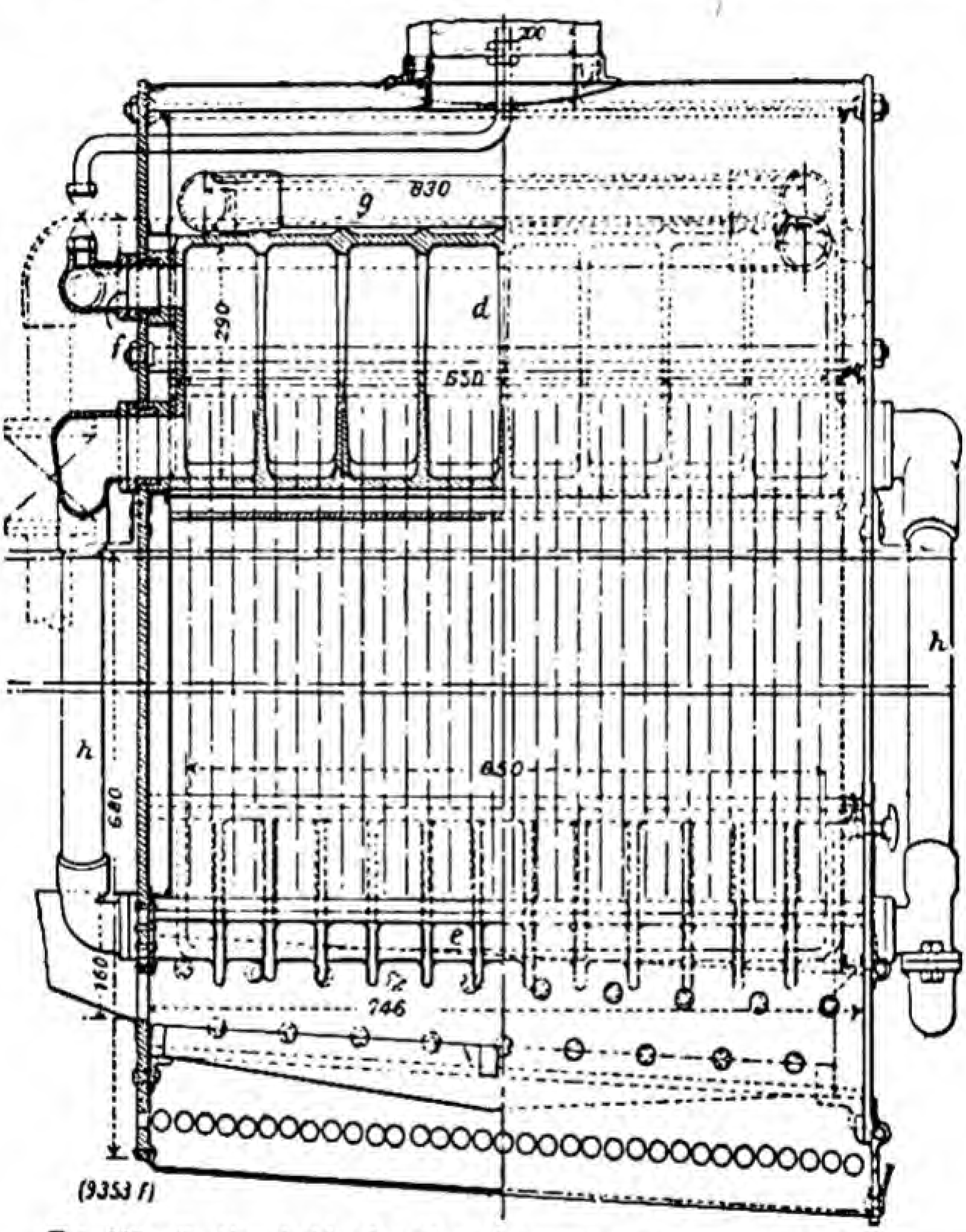


FIG. 259.—Sectional side elevation of steam waggon water-tube boiler.

furnace gases and serve to dry the steam. The steam on leaving *g* passes through a valve to the engines. The hot furnace gases pass over practically the whole surface of the boiler. The fuel is fed into a hopper and is jolted down the inclined plane of the bars by the motion of the waggon. A casing with air spaces encloses the whole boiler, the internal portion being protected by asbestos and sheet iron. The feed arrangements will be explained later.

The engines. The use of two independent sets of engines, each connected to one of the driving wheels, obviates the necessity for

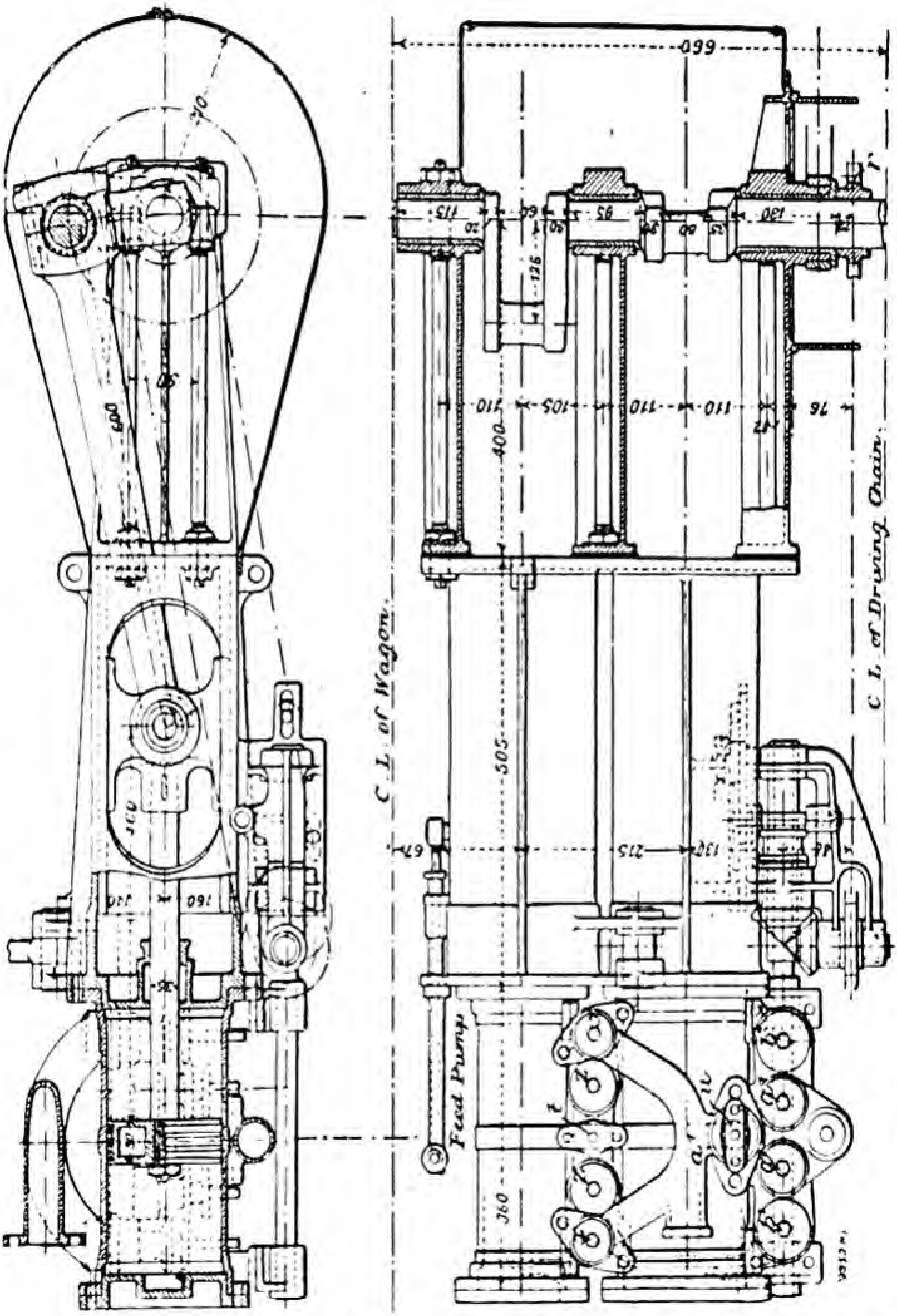


FIG. 260.—Arrangement of one of the compound engines for Barber's steam waggon.

complicated speed changing and differential gearing. Each set has two cylinders and can be arranged to work as a compound engine or as a simple engine having high pressure steam admitted to both

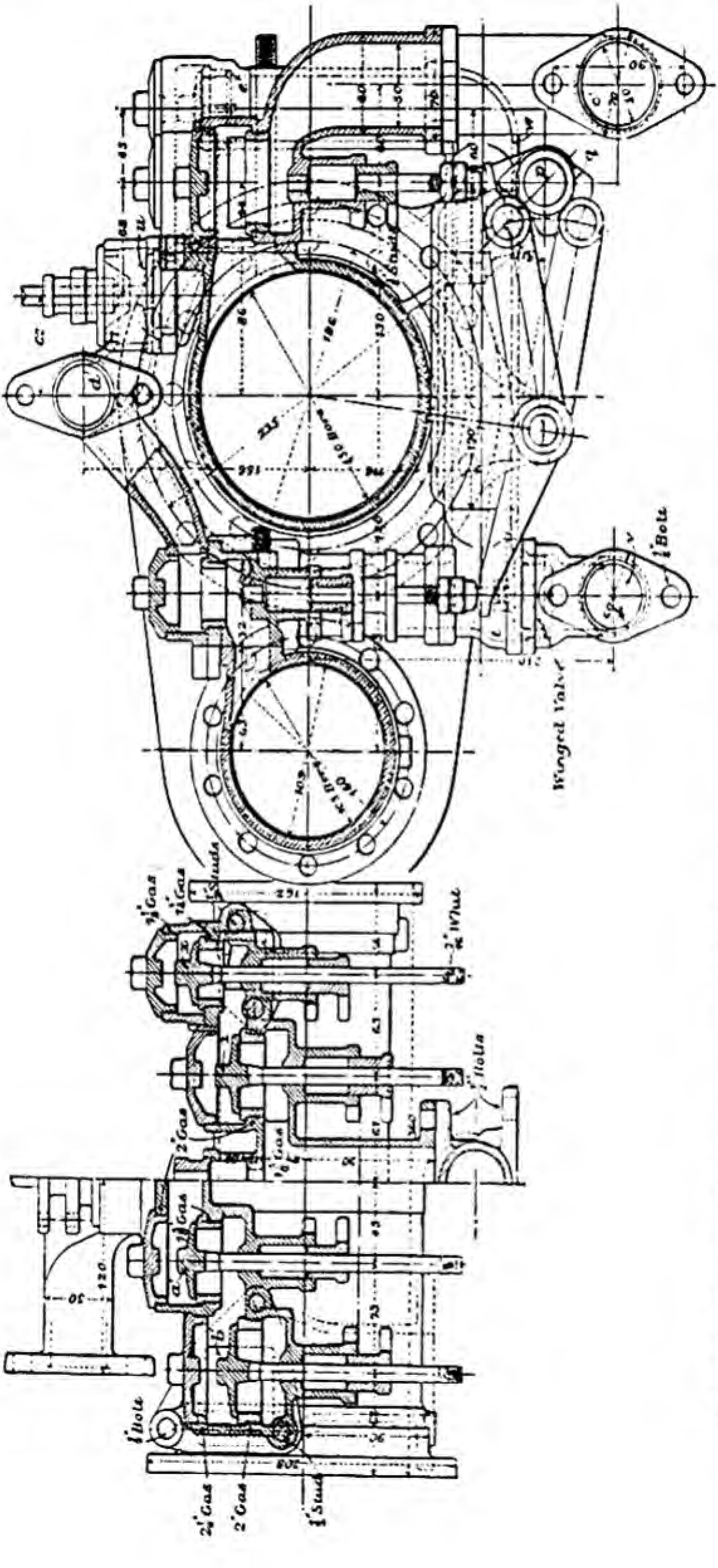


FIG. 261.—Longitudinal section of the valve box and cross section of the cylinders in the steam waggon engines.

cylinders. The general arrangement is shown in Fig. 260. When running compound, the valves t and u are closed, the former shutting off the exhaust to the condenser from the high pressure cylinder, and the latter the live steam from the low pressure cylinder. The exhaust from the high pressure cylinder then passes along the pipe w into the low pressure cylinder through the valve e' (Fig. 261); the exhaust from the low pressure cylinder

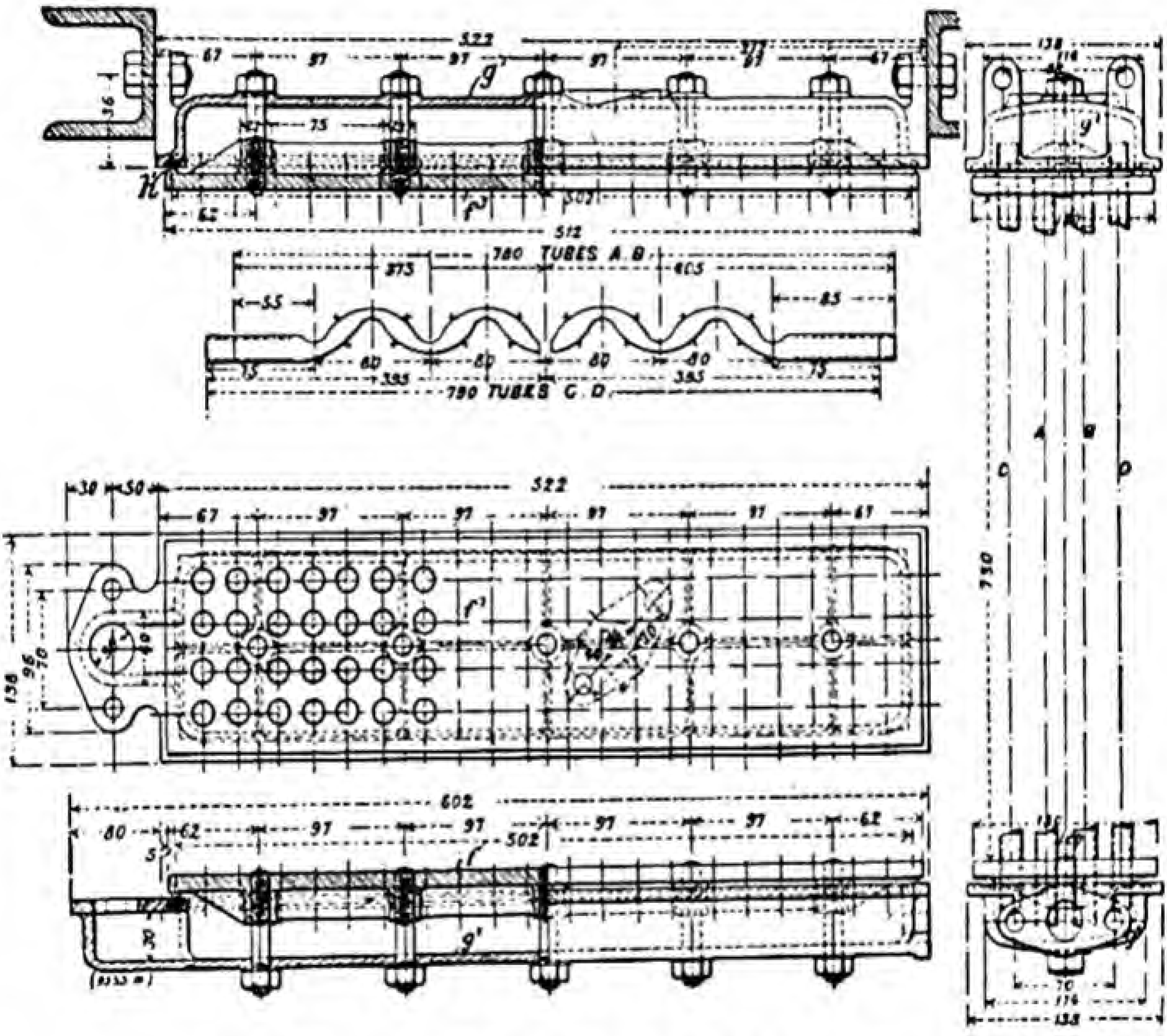


FIG. 262.—Details of surface condenser for steam waggon.

is led through a pipe c' to the condenser. In running as a simple engine, both t and u are open and steam enters the high pressure and low pressure cylinders through the valves x and a' respectively (Fig. 260); exhaust takes place through the valves z and b', and thence along the pipes v and c' (Fig. 261) to the condensers.

The arrangement enables a great range of power to be obtained simply. The engines are slung from the waggon frame by means of links and are kept in correct position relative to the rear axle by means of radius rods, which are adjustable, to provide for taking

up the chain. The crank shafts are of nickel steel and run in an enclosed oil bath.

The valves for distributing the steam are of the simple drop type (Fig. 261). These are actuated by means of cams on the shaft p, which is driven by a Renold chain from a sprocket wheel on the crank shaft.

The details of the condenser are shown in Fig. 262. It consists of two chambers connected by brass tubes expanded into the tube plates. The tubes are of similar design to those of the boiler, and are kept cool by an air draught produced by a fan.

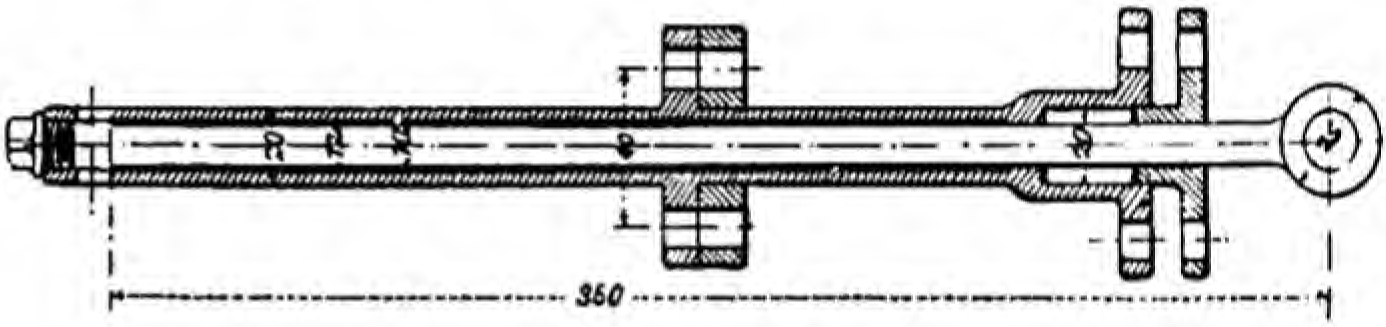


FIG. 263.—Steam waggon feed pump.

Feed arrangements. The feed water is drawn from the condenser and fed into the boiler by two feed pumps (Fig. 263), one driven from each high pressure crosshead pin. The supply of water to

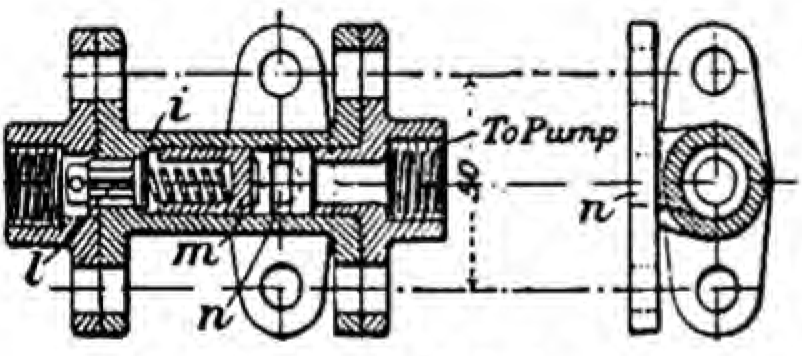


FIG. 264.—Automatic feed valve.

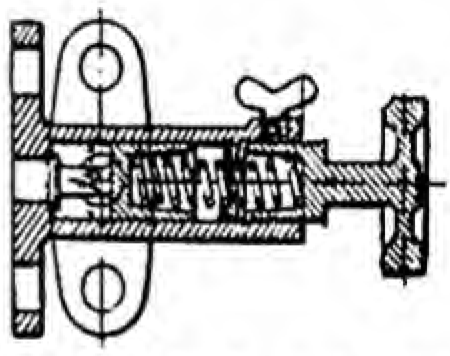


FIG. 265.—Feed relief valve.

the boiler is regulated automatically by the following arrangement. An automatic valve is shown in Fig. 264, consisting of a cylinder i having a piston m, which may move freely in the cylinder under the control of a helical spring. The pump delivery pipe is connected to one end of the cylinder; at the other end is a small check valve l and a pipe leading to the boiler at normal working level. A branch connection n leads the feed water to the boiler. The orifice of n is open or closed, depending on the position of m in the cylinder. The pump reciprocates m, pushing it out of the way

during each delivery stroke, the spring returning *m* during the suction stroke and so closes *n*. This action takes place freely should the water in the boiler be below normal working level; when this is the case steam passes from the boiler through the check valve *l*, and being compressible, permits of the reciprocation of *m*. Should the water level in the boiler be higher, water will enter the automatic valve, and, being practically incompressible, will prevent the reciprocation of *m* and thus shut off the feed delivery. In this event, the water discharged from the pump escapes through a feed relief valve (Fig. 265) on the pump delivery pipe and returns to the condenser. A small hole is provided in the check valve *l* to permit of the gradual return to the boiler of the water behind *m* as the level falls again.

EXERCISES ON CHAPTER XVI.

1. Give an outline sketch showing the piston and its connection to the driving axle of a locomotive. What is the object in having four or more driving wheels coupled together?

2. Sketch in section a locomotive cylinder and describe its construction. Omit the slide valve.

3. Sketch and describe a locomotive piston, piston rod and crosshead.

4. Explain the construction and give sketches of the big-end of a locomotive connecting rod.

5. Sketch the arrangement for reversing any locomotive you know. Name the parts in your sketch.

6. Sketch and describe a locomotive driving axle and one of the driving wheels. Show clearly how the tyre is secured.

7. Sketch and describe Holden's injector for liquid fuel.

8. Give sketches and description of a steam boiler adapted for use in a road locomotive such as a steam lorry or a motor car.

9. Sketch with as little detail as possible the arrangement of a steam engine suitable for any road vehicle other than a traction engine.

10. A locomotive develops 250 horse-power at a speed of 50 miles per hour. What is the average pull exerted on the train?

11. What would be the indicated horse-power of a locomotive when moving at a steady rate of 35 miles per hour on a level rail, the weight of the train being 130 tons and the resistance to traction 10 lbs. per ton? 1896.

12. If a locomotive of 1200 I.H.P. uses 38 lbs. of feed water per hour per I.H.P.; in a journey of $2\frac{1}{4}$ hours what is the total amount of feed water? If every pound of coal produces 9 lbs. of steam, what is the total weight of coal burnt on the journey? If the mechanical efficiency of the engine is 0·85, what is the power actually spent in overcoming the resistance of the engine and train? 1904.

CHAPTER XVII.

THE STEAM TURBINE.

The steam turbine.—Although the earliest recorded steam engine (dated about 2000 years ago) was a turbine in principle, it is only within recent years that the machine has been made successful commercially, and there is still a great deal of experimental work to be done in connection with it. There are no reciprocating parts in a turbine. The fluid delivers up its energy to revolving wheels in flowing past blades or buckets fixed to the rims, being guided properly by fixed blades, so as to come into contact with the moving blades with as little mechanical shock as possible.

Water turbines have been applied successfully for many years, and their theory is generally understood. In turbines working with steam or other elastic fluids, the principles governing the action are affected by the property of expansibility which the fluids possess.

There are two types of turbines:

(*a*) Those in which the energy of the fluid when it enters the wheel is practically entirely in the **kinetic form**;

(*b*) Those in which the energy of the entering fluid is chiefly of the **pressure form**.

Turbines of the first-mentioned type are called **impulse turbines**: those of the latter type are called **reaction turbines.** The de Laval turbine is an example of the impulse turbine; the Parsons turbine is of the reaction type.

The de Laval steam turbine.—This steam turbine is manufactured in this country by Messrs. Greenwood & Batley, Ltd., to

whom the author is indebted for the illustrations. The action of the turbine may be understood by reference to Fig. 266. Steam is blown through a number of nozzles and is directed against the vanes of a single revolving wheel. The wheel is contained in an outer casing not shown in this illustration; the steam, after passing the wheel is exhausted from the casing to the condenser or to the atmosphere. One revolving wheel only is used in this machine.

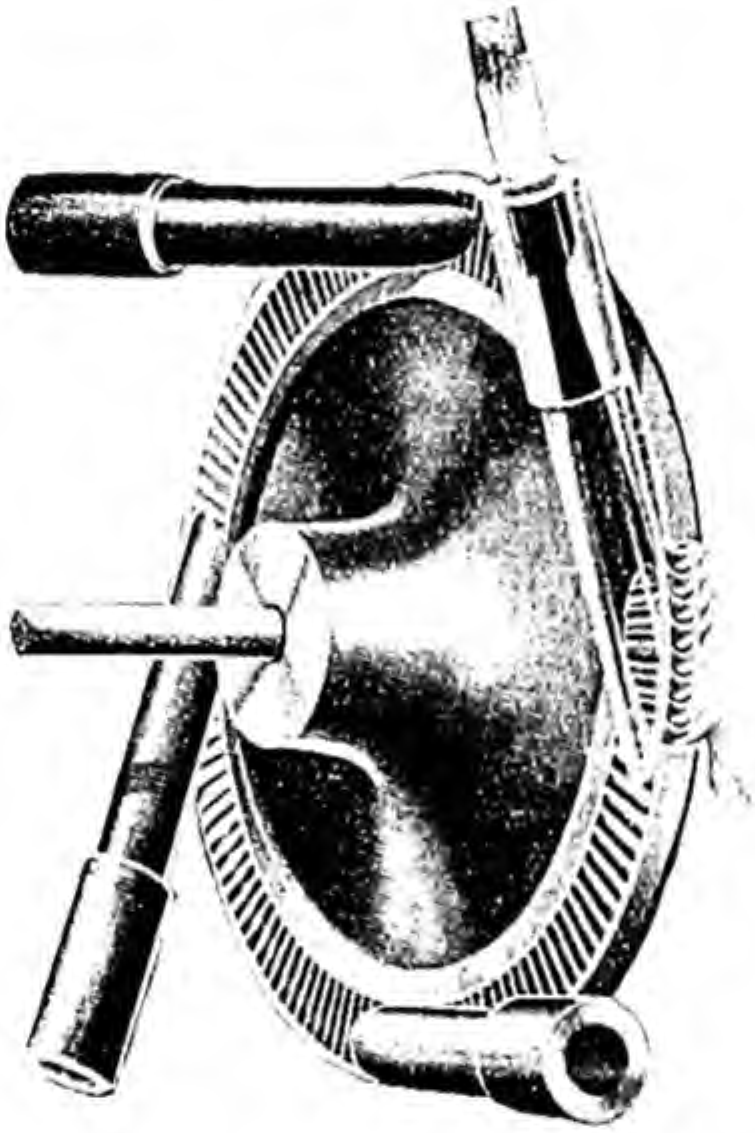


FIG. 266.—Action of the de Laval steam turbine.

The nozzle is so constructed as to permit the steam passing through it to be expanded completely before reaching the wheel. The pressure of the steam at the mouth of the orifice is thus very low, its volume per pound weight is very high and its velocity is also very high. In fact, the function of the nozzle is to convert, so far as is possible, the pressure energy of the steam delivered to it into a corresponding amount of kinetic energy, the latter kind of energy, it will be remembered, being proportional to the square of the velocity.

Owing to the very short time that the steam is actually in the nozzle, practically no heat is wasted, *i.e.* the flow through the nozzle is adiabatic, consequently the conversion of energy is practically perfect. It has been said that the velocity of the steam is very high on discharge from the wheel, and it follows from this that the velocity of the wheel is also very high. In practice, the wheel rims of the de Laval turbines run at from about 500 to 1400 feet per second, giving, for the 5 H.P. size a speed of revolution of 30,000 per minute and for the 300 H.P. size 10,600 revolutions per minute.

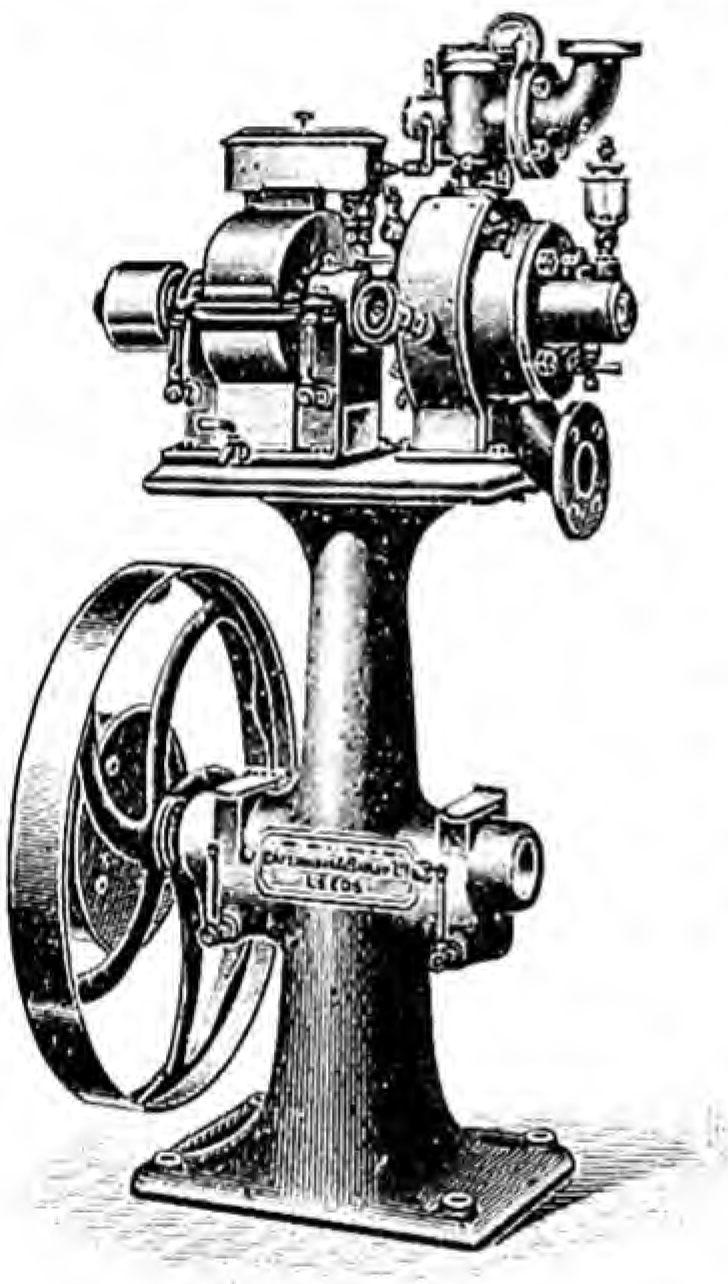


FIG. 267.—3 B.H.P. de Laval steam turbine.

Such speeds of revolution are generally much too high to be utilised directly, and the speed of the wheel shaft is reduced by means of gearing. Fig. 267 shows a complete machine of 3 B.H.P., the separate parts being shown in Fig. 268. Referring to the latter, A is the wheel shaft carrying the turbine wheel B, which is supplied with steam through the nozzle I fitted with a valve K for shutting off steam. In the larger turbines having several nozzles, each is fitted with a shut-off valve (Fig. 269), so that the steam supply may be adjusted by hand, independently of the governor, to suit the work being done. The valve is omitted in the smaller sizes of machine having one nozzle only. The energy supplied is adjusted to meet the demand by closing entirely one or more of the nozzles.

The shaft A carries a double helical-toothed pinion C (Fig. 268) gearing with a similar wheel M mounted on a second gear wheel shaft N. In the turbine supplied to the Engineering Laboratory at the West Ham Technical Institute the ratio of M to C is 8 to 1,

and the turbine wheel runs at 30,000 revolutions per minute; consequently N runs at 3750 revolutions per minute. Further

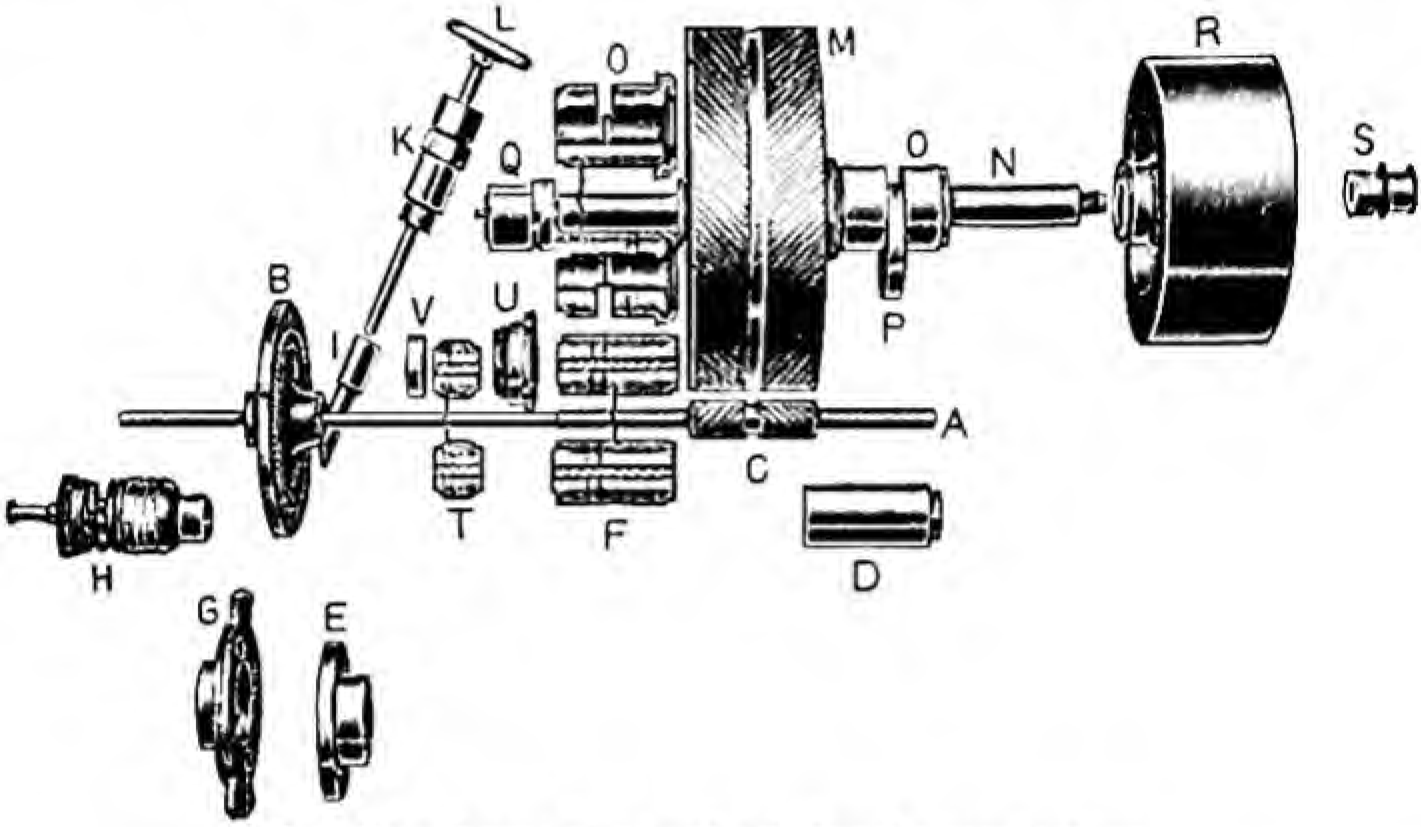


FIG. 268.—Separate parts of a de Laval steam turbine.

reduction, if required, is secured by means of a belt pulley R mounted on N and connected to a belt pulley on a third shaft, which may be observed in Fig. 267, the bearings being formed in the pedestal of the machine.

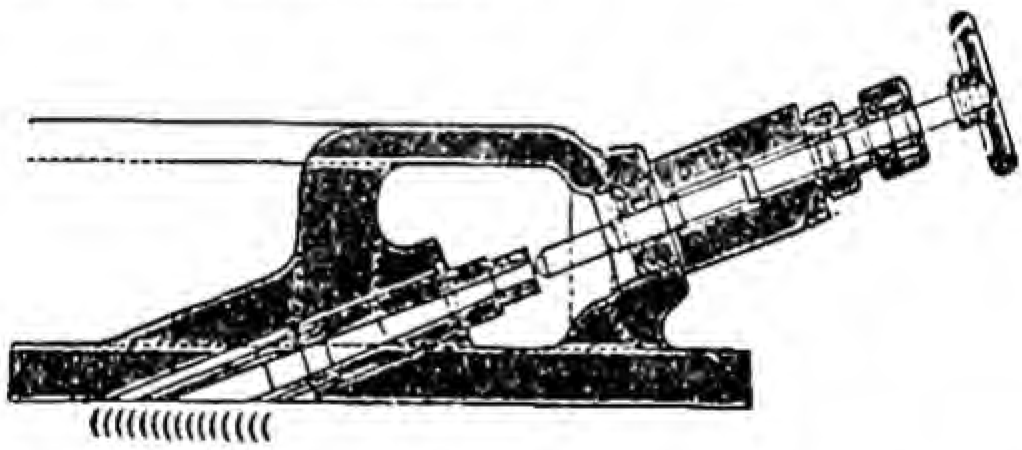


FIG. 269.—Nozzle and shut-off valve of the de Laval steam turbine.

The speed of the turbine wheel is kept uniform by means of a centrifugal governor Q (Fig. 268) mounted on the second gear shaft, and operating a throttle valve in the steam admission pipe. The governor is shown in detail in Fig. 270 and diagrammatically in Fig. 271. The construction is adapted to the high speed at which the governor runs. The gear shaft is shown at A (Fig. 271) and has a collar B mounted on it, which serves to pivot two

bent levers C, C. These levers are pivoted at D, D, and bear at E against the end of the throttle valve spindle F, which is loaded by means of a spring G. This spring tends to keep the throttle valve open. During running, the masses of the bent levers, indicated by the balls shown in Fig. 271, undergo centrifugal force tending to carry them outwards from the spindle. The motion is communicated to the point E, thus putting the spring G under compression and partially closing the throttle valve. In the actual governor (Fig. 270) the bent levers consist of the halves of a split collar, which serve to enclose the remainder of the working parts of the governor. The governor is very efficient, the variation of speed generally not exceeding 2 or 3 per cent. between full load and no load. The construction of the throttle

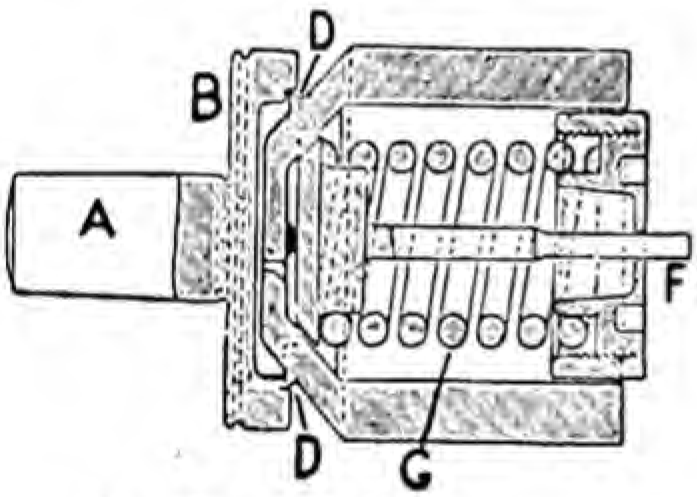


FIG. 270.—de Laval governor.

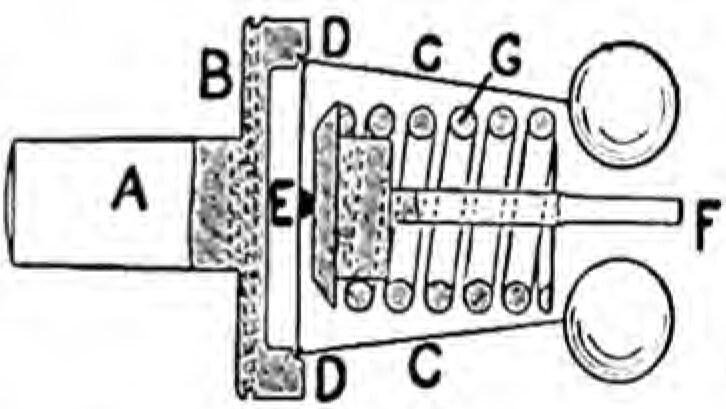


FIG. 271.—Diagram of the de Laval governor.

valve, which is of the double valve equilibrium type, may be understood by reference to Fig. 272, where the valve is shown at D; dirt is prevented from entering the turbine by a wire gauze cage C through which the entering steam passes.

General arrangement of the turbine.—Fig. 272 shows the general arrangement of a 20 H.P. de Laval turbine. On leaving the throttle valve D the steam enters the casing E and after passing through the wheel F is discharged into R and exhausted through N. The gear wheels H and J are enclosed in a case, which forms an oil bath necessary for smooth running.

Construction of the wheel.—Owing to the manner in which the steam is used, there is no necessity for a small clearance between the wheel and the casing; the wheel does not touch anywhere, and runs perfectly freely. There is about $\frac{1}{16}''$ clearance between the mouth of the nozzle and the wheel blades. Special attention

has to be given to the construction of the wheel and its shaft on account of the high speed. The wheel is a solid disc, on the

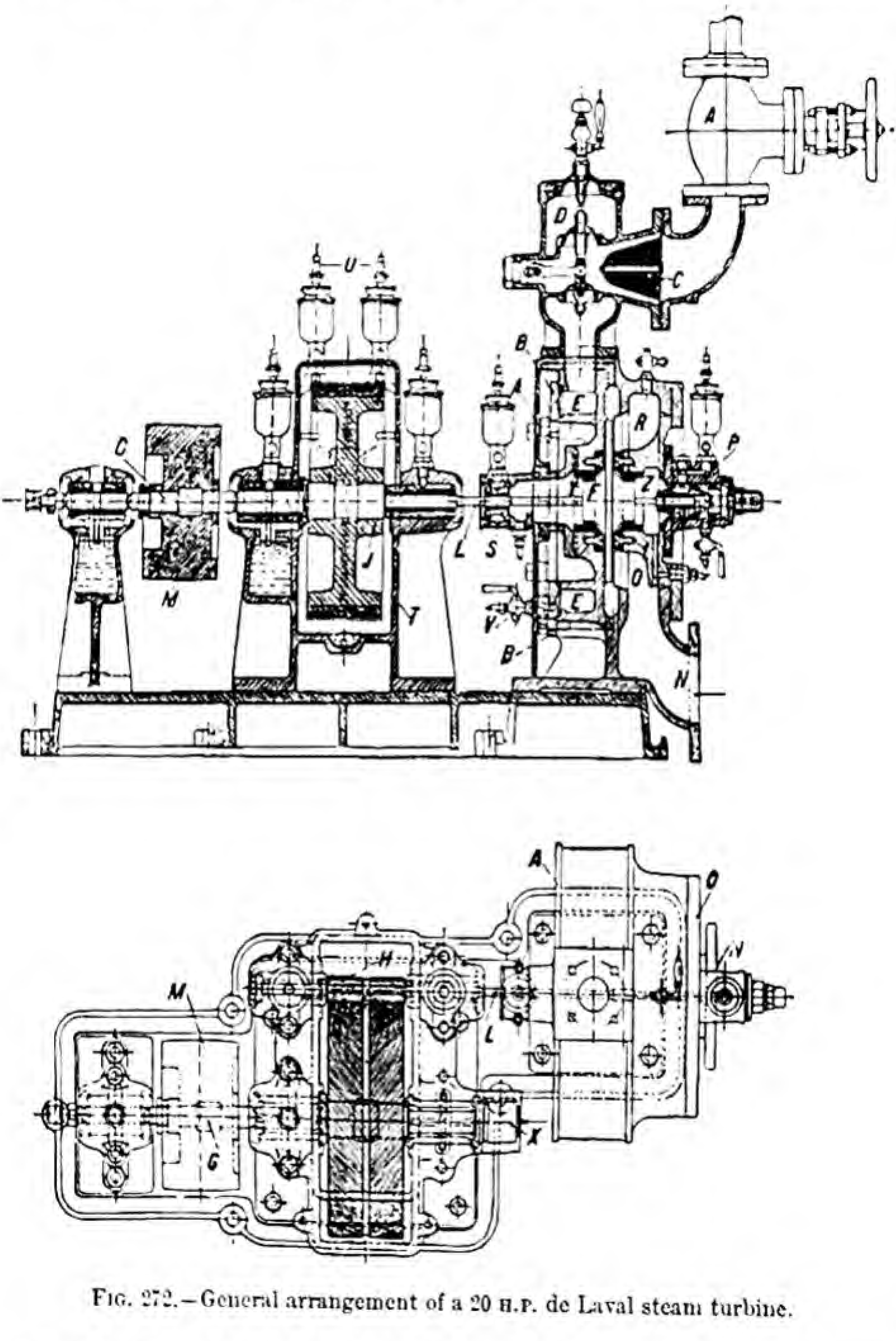


FIG. 272.—General arrangement of a 20 H.P. de Laval steam turbine.

circumference of which the buckets are fixed, each bucket being made and dovetailed separately to the wheel. It may be mentioned

that the centrifugal force on the bucket of a 300 H.P. turbine wheel, the weight of the bucket being 250 grains, is 15 cwts. when the wheel is running at its normal speed. The construction of the wheel will be understood by reference to Fig. 273, in which also the buckets are shown separately enlarged in section.

No matter how carefully the wheel may be turned and balanced, it is impossible completely to get rid of centrifugal forces which would, unless precautions were taken, entirely ruin the bearings in a few minutes. The plan which is adopted is to mount the wheel on a flexible shaft of small diameter having bearings on each side of the wheel, but at a good distance from it. On rotation commencing, the centrifugal effects due to want of perfect balance produce vibrations; the vibrations, however, entirely disappear at a certain speed and do not reappear. This takes place through the wheel being permitted, by the flexibility of the shaft, to find its natural axis of rotation in much the same manner as a top spinning on the floor. The shaft of the 150 H.P. turbine is about 1″ diameter. This shaft transmits the whole power of the turbine wheel to the gearing, but, owing to the speed of transmission, the turning moment is low, and consequently the stresses in the shaft do not exceed a safe limit. The gear wheel pinion is made of hard steel in one piece with the flexible shaft; the teeth of the gear wheel are of somewhat softer steel. The teeth are machine cut, double helical, of small pitch, and run very quietly. The bearings of the flexible shaft are lubricated by sight feed lubricators; those of the slow speed shaft by means of ring lubricators. All these bearings are lined with white metal.

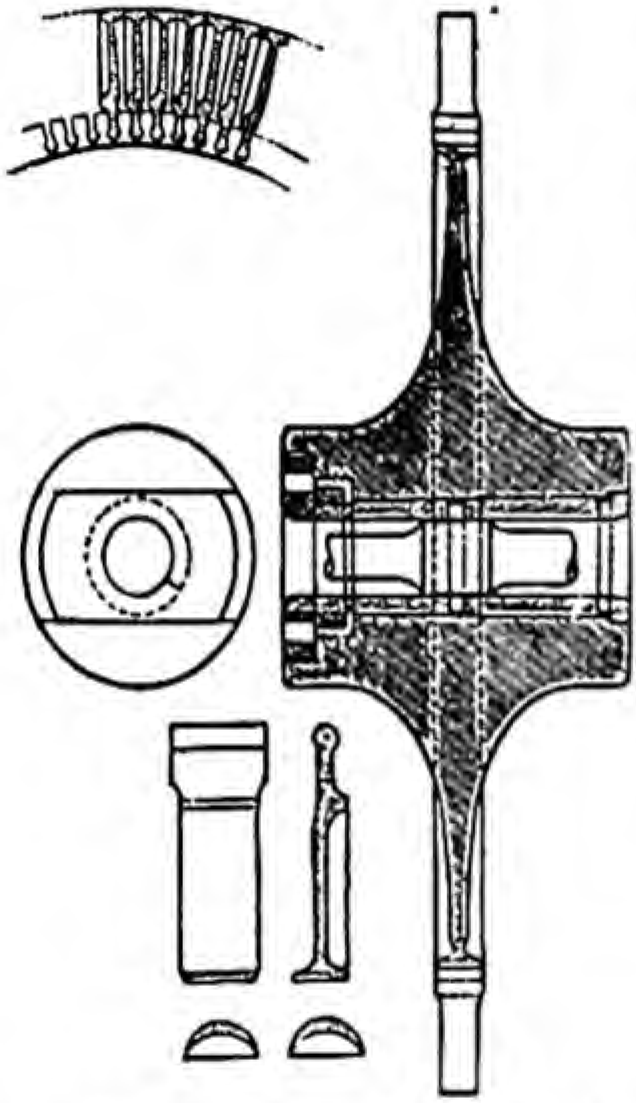


FIG. 273.—Construction of the wheel and buckets of a de Laval steam turbine.

Losses in the machine.—As there is no leakage of steam (the

whole of which passes through the wheel buckets), and very little friction due to the steam rubbing on the buckets, the only losses of importance are due to the residual kinetic energy of the leaving steam. The resistances of the moving parts are made up of two parts, viz. the friction of the bearings which is small, and the resistance of the contents of the casing in which the wheel revolves. In turbines discharging direct to the atmosphere, the latter resistance may be high, and is considerably reduced by rarefying the contents of the casing by use of a condenser and air pump, or of an ejector condenser for the production of a vacuum. The efficiency of the machine is therefore greatly increased by the employment of a good vacuum. With 28″ vacuum, the total resistances of the machine amount to about 7½ per cent. in the larger sizes of turbines, giving a mechanical efficiency of 92½ per cent.

Applications of the de Laval turbine.—The turbine may be used for driving electric generators, the shaft of the dynamo being directly connected to the gear shaft, thus saving belt and countershaft connections as well as floor space. Blowers and fans may be connected in a similar manner as such machines can utilise efficiently the high speed of rotation. An interesting application is in the driving in this manner of centrifugal pumps for forcing water at high pressures, up to 250 or 300 lbs. per square inch.

RESULTS OF TRIALS WITH A DE LAVAL HIGH PRESSURE TURBINE PUMP.

	Height of delivery, feet.	Gallons of water delivered per min.	H.P. imparted to water.	Lbs. of steam per water H.P. per hour.	Remarks.
1st Trial, - -	312	529	50	34·2	Work for condensing is included.
2nd Trial, - -	443	629	84·4	29·0	
3rd Trial, - -	509	529	81·6	31·3	
4th Trial, - -	467½	640	90·7	30·6	
		530	75·0	32·2	
		430	60·9	35·1	
		286	40·5	43·1	

Test on a de Laval turbine.—The foregoing results are taken from a paper by Mr. Konrad Anderson and refer to a set of tests made on a steam turbine driven plant in which two pumps were used, one centrifugal pump being driven by the turbine by means of gearing at a low speed and delivering water to a high speed centrifugal pump driven direct from the steam turbine, which in turn forces the water to a considerable head.

In these tests there was a considerable distance between the boiler and turbine, giving a probability of wet steam. The condensing water was supplied from the slow-speed pump, consequently the machine drove its own condenser.

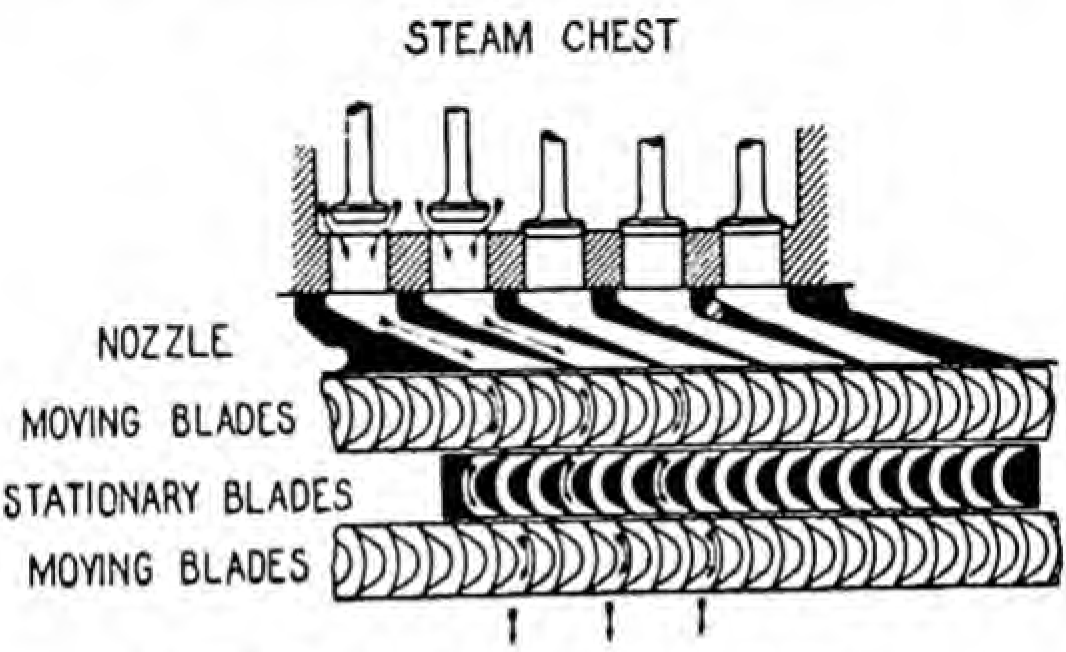


FIG. 274.—Action in the Curtis steam turbine.

The Curtis steam turbine.—The Curtis turbine is made in this country by the British Thomson-Houston Co., Ltd., the illustrations given here being reproduced by the courtesy of this firm. The turbine is of the "impulse" or "velocity" type, in which a high velocity is given to the steam by expanding it in stages through nozzles, so as to convert its pressure energy into the kinetic form. After leaving the nozzles, the steam passes successively through two or more rings of buckets on a revolving wheel interspersed with reversed buckets on the stationary part of the turbine. After leaving this wheel the steam is redirected and further expanded by a second ring of stationary nozzles to be passed through a second revolving wheel and so on. The arrangement of the first revolving wheel and nozzles is shown in diagram form in Fig. 274. Each revolving wheel is arranged to abstract a portion only of the kinetic energy of the steam passing through it, and

thus may revolve at a much lower speed than in that type of turbine in which the whole operation is carried out by the use of one revolving wheel only. The expansion of the steam is effected in stages, *i.e.* is not completed until the last set of nozzles has been passed. Each set of guide nozzles and revolving wheel is called a **stage**, a turbine having four revolving wheels being designated a **four stage turbine**. The number of stages and the number of rings of buckets in each stage is governed by the degree of expansion, by the peripheral speed of bucket which is desirable, and by other conditions of mechanical expediency.

The governing in this turbine is effected by means of a group of admission valves (Fig. 274) controlling the admission of steam to each of the first set of nozzles. The valves are under the control of the governor so that the number of valves open, and consequently the number of nozzles in operation, will be proportional to the load on the turbine.

The turbine, as made by the above mentioned firm, is generally applied to the direct driving of electric machinery, and the most convenient arrangement has been found to be that in which a vertical revolving shaft is employed. Fig. 275 shows a complete four stage machine in section. The revolving shaft A has the turbine elements mounted at B. C is an electric generator mounted on a part of the shaft A direct coupled. A surface condenser is shown at D.

Steam is supplied to the turbine through the admission valve V, and first set of nozzles N_1. The four revolving wheels, W_1, W_2, W_3, W_4, are fixed to the shaft A and are separated by diaphragms arranged so as to prevent steam leaking from stage to stage except by passing through the nozzles N_2, N_3, N_4, which are formed in the diaphragms. The steam is subsequently discharged into the surface condenser D, in which the vacuum is maintained by an air pump independently driven by an electro-motor.

The shaft A is mounted on a footstep bearing which introduces so little friction that the rotating parts of a 1500 kilowatt set can be easily turned round by hand. The footstep bearing consists of a bearing block F which rotates with the shaft, and another G which is fixed. Water is used as a lubricant, being forced through a central hole in the stationary block and spread outwards as a thin film between the surfaces of the blocks. The water is then passed

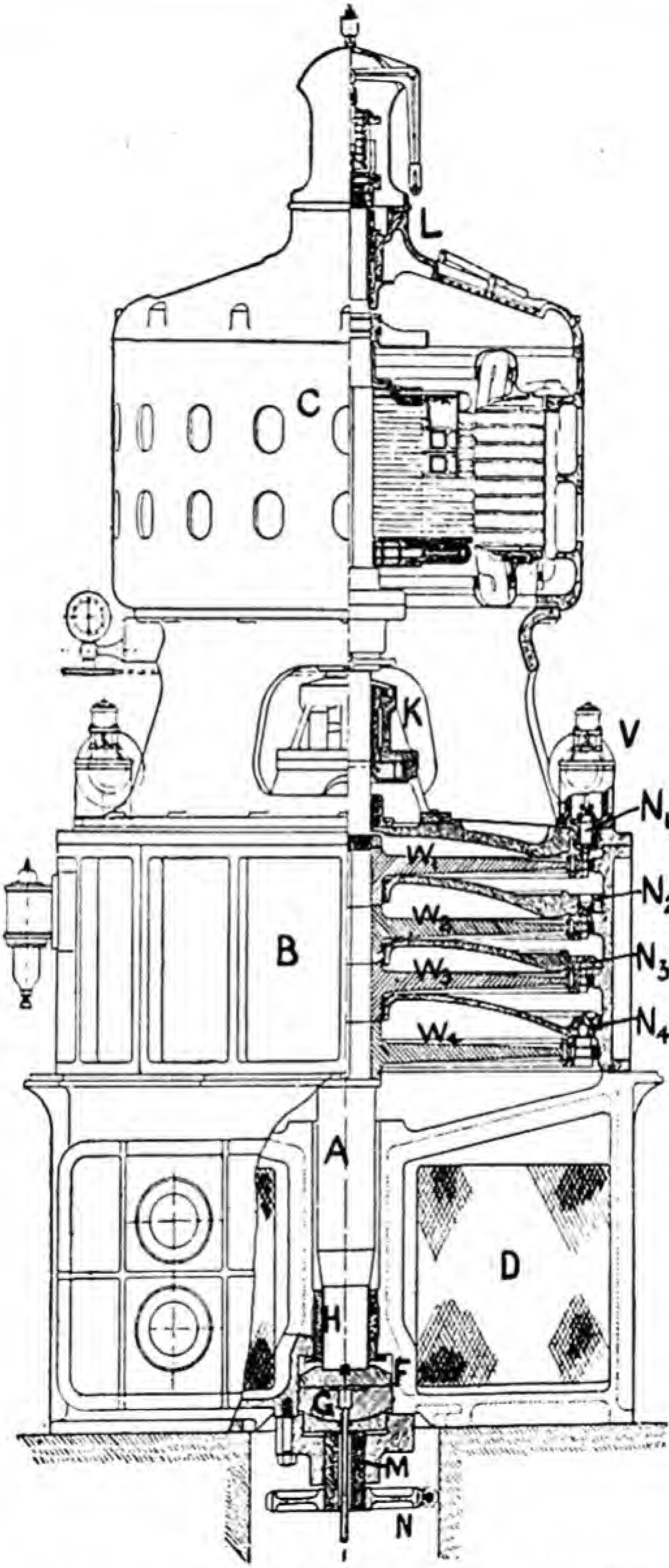


Fig. 275.—Sectional elevation of a Curtis steam turbine and electric generator

upwards and lubricates a guide bearing *H*, afterwards passing into the turbine base to be removed with the condensed steam. The arrangement obviates the necessity for a packing gland at this part of the turbine. To maintain the water pressure to the footstep bearing, a small pressure pump and hydraulic accumulator are used; the power thus employed is insignificantly small. It will be noticed that, by this arrangement, the metal surfaces of the footstep blocks are never in contact during working; there is always a film of water between them, and the friction to be overcome is consequently that of metal rubbing on water.

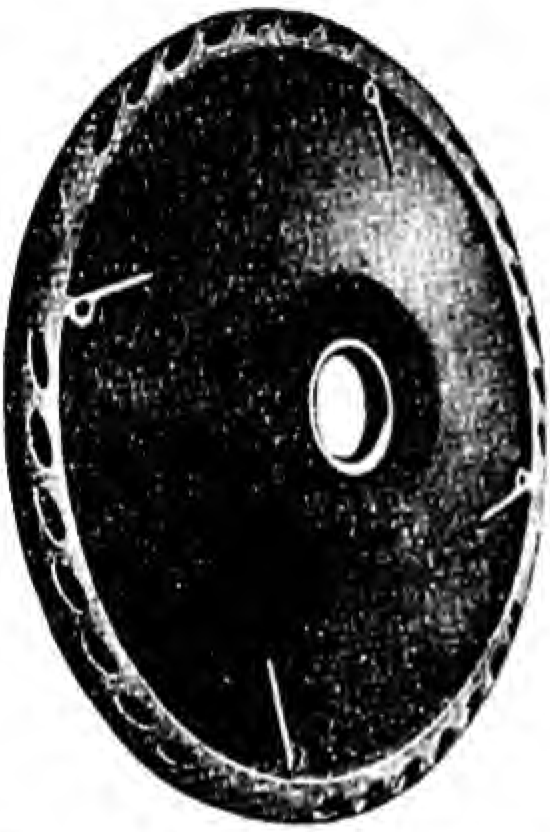


FIG. 276.—Diaphragm and nozzles of the Curtis steam turbine.

FIG. 277.—Curtis bucket ring.

The middle bearing *K* and the top bearing *L* are oil lubricated, the oil being delivered to them from a small pump driven from the shaft of the water pressure pump, and returned to be used over and over again. The middle bearing *K* is provided with a packing gland to render the shaft *A* steam- and air-tight where it passes through the upper cover of the turbine casing.

The construction of the diaphragms and nozzles will be understood by reference to Fig. 276. One of the rings of the revolving buckets is shown in Fig. 277. The buckets are cut out of the solid rim by special machinery and are shrouded by a steel rim (Fig. 278). Loosening of the buckets at the high speeds of revolution employed is thus impossible. The clearance between

the revolving wheels, the intermediate buckets, and the diaphragm nozzles must be very small, and must be adjustable in order to prevent actual contact taking place. This is managed by a screw adjustment on the footstep bearing *M* (Fig. 275), operated by means of a worm wheel and worm at *N*. The shaft *A* and its attachments may thus be raised or lowered for adjusting the clearance. The complete set of revolving wheels for a four stage turbine is shown in Fig. 279. It will be noticed that each wheel

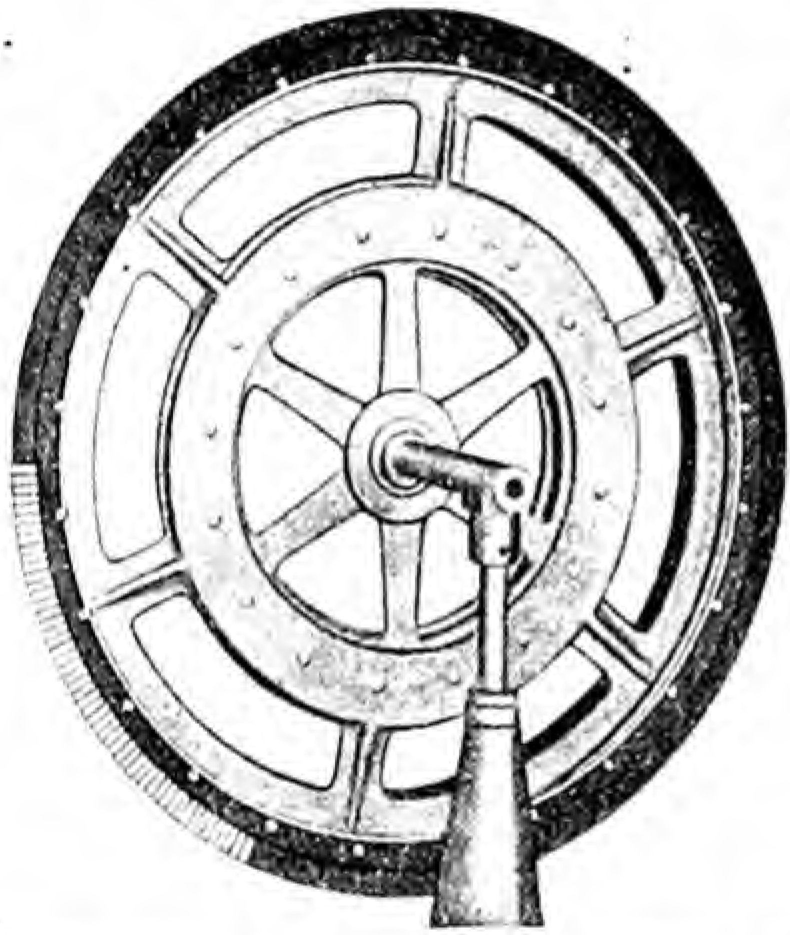


FIG. 278.—Wheel with buckets partly cut.

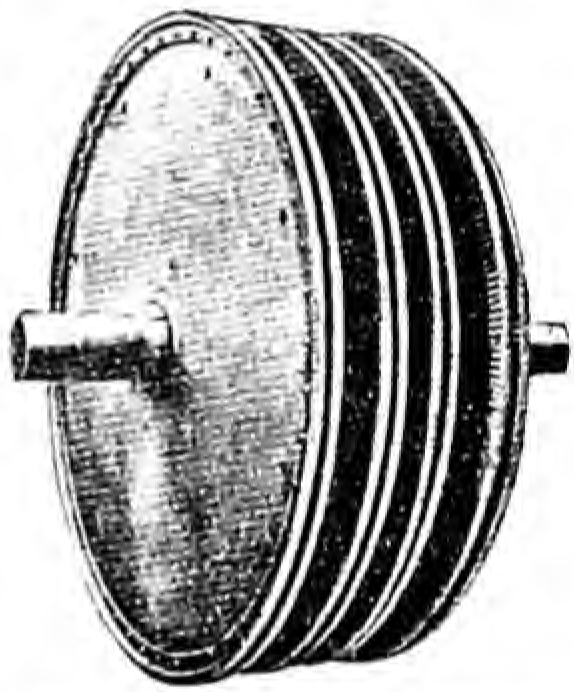


FIG. 279.—Set of wheels for a four-stage Curtis steam turbine.

has two rings of buckets; the intermediate guide buckets on the casing being placed between these rings (Fig. 275).

The governor is of the centrifugal spring loaded type, and is connected to an electric controller, which opens or closes the circuits of a series of electro-magnets. These magnets operate pilot valves, which in turn open or close the steam admission valves of the first set of nozzles. The valves are of the balanced type and thus require but little effort to operate. The principle followed in the whole arrangement is to have the governor perform as little mechanical work as possible, thus increasing its sensitiveness.

This type of turbine may be conveniently applied to work with superheated steam. As the greater part of the heat is converted

into kinetic energy in the first set of nozzles, the buckets are not subjected to inconveniently high temperatures. The floor space occupied is very small. Fig. 280 shows the power station at the maker's works; the Curtis turbo-generator shown to the left of the illustration is equal in output to the aggregate capacity of the three reciprocating engine sets.

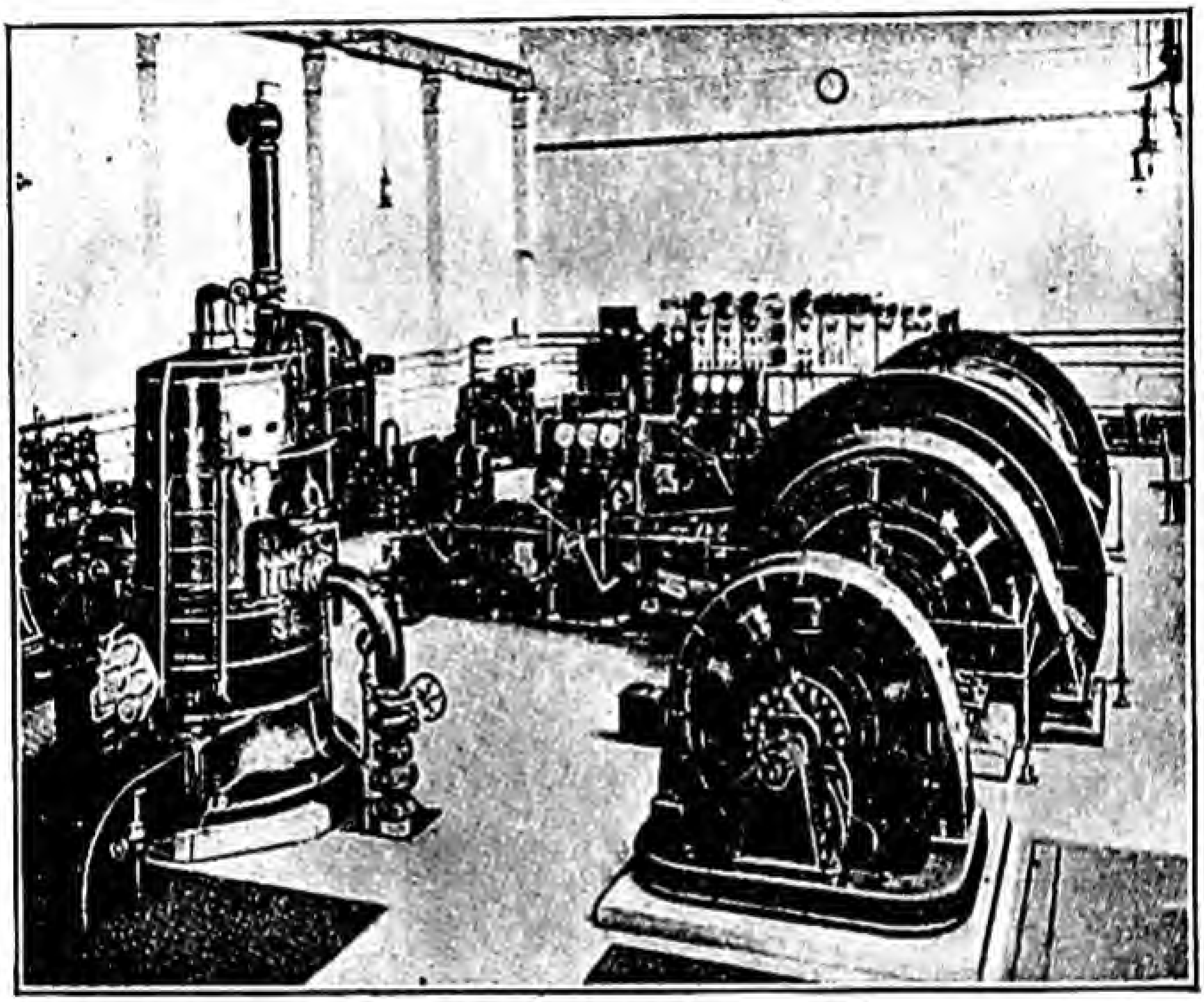


FIG. 280.—Power station at the British Thomson-Houston Works, showing a Curtis turbo-generator at the left of the illustration.

The Parsons steam turbine.—By the courtesy of Messrs. Willans & Robinson, illustrations are given here of the turbine manufactured by them under license from the Hon. Charles A. Parsons, C.B. The makers have introduced several important features of construction which differentiate their machine from others of this type, but the main principle is that of the Parsons parallel flow turbine.

An external view of the 1000 kilowatt turbine is shown in Fig. 281, and the same machine with the top cover of the casing opened back in Fig. 282. It will be noticed from the latter illustration that the turbine consists of a rotating shaft having a large

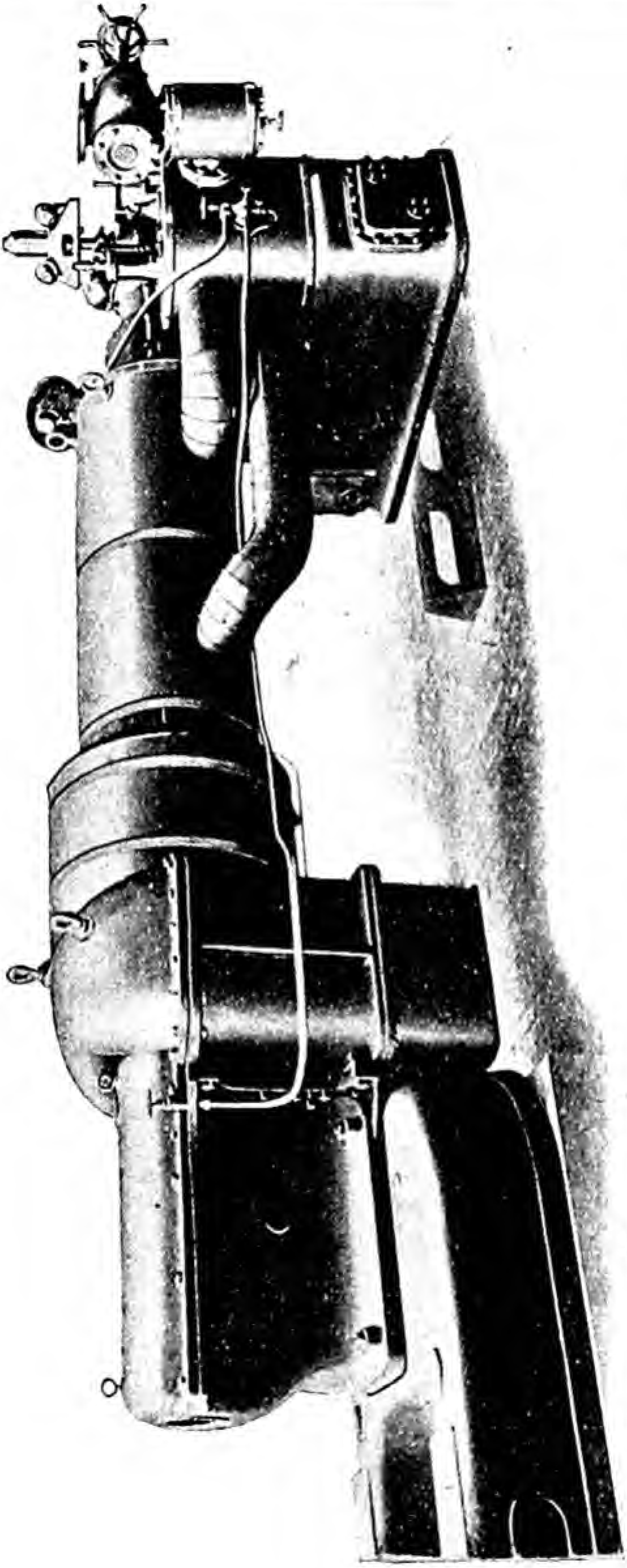


Fig. 281.—External view of 1000 kilowatt steam turbine on the Parsons system; constructed by Messrs. Willans & Robinson.

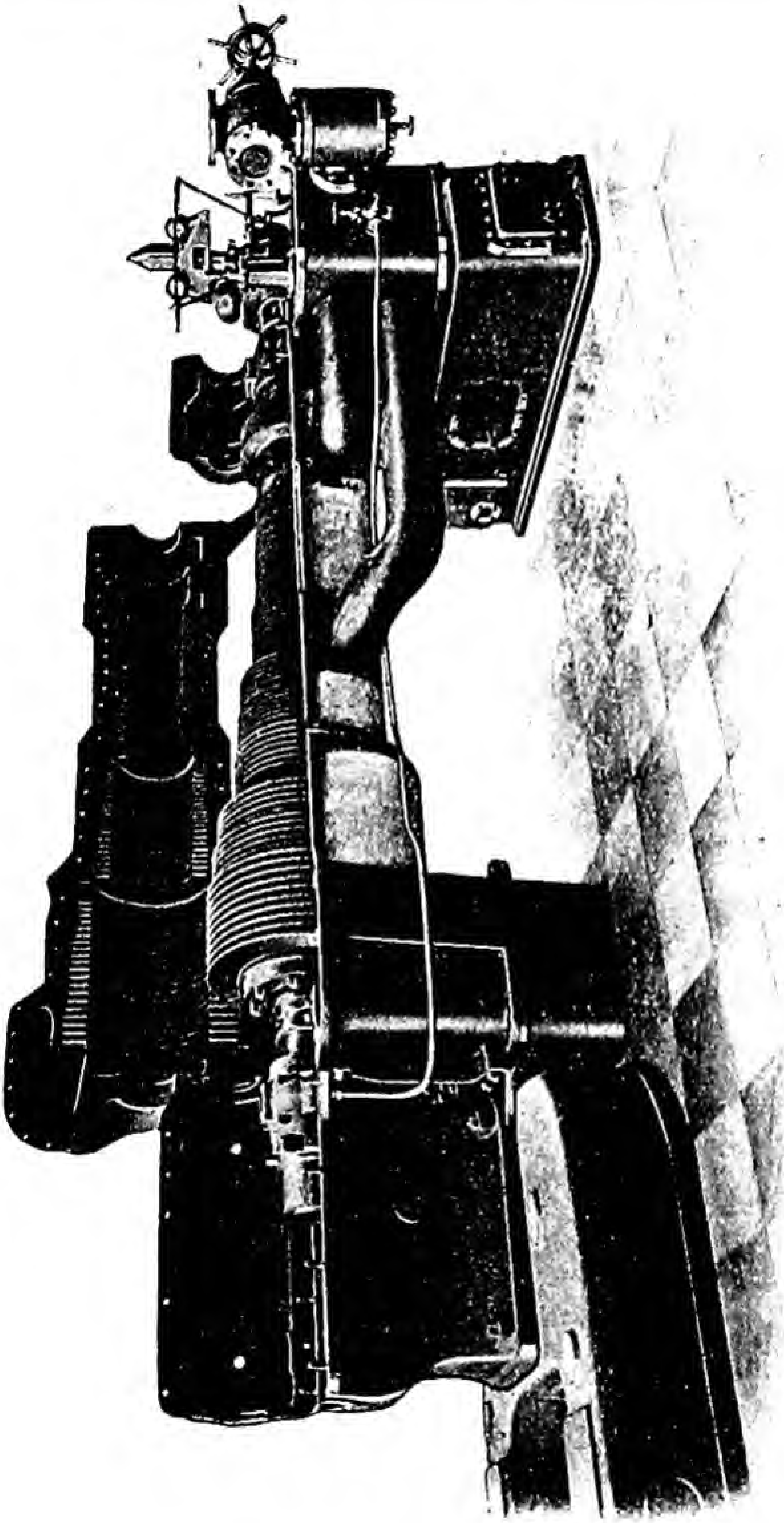


FIG. 282.—The turbine of Fig. 281, with the top cover opened back.

number of bladed rings which are of increasing diameter, corresponding bladed rings being attached to the casing so as to come between the rotating rings. Steam enters the turbine near the smaller end and flows among the blades towards the larger end, expanding and falling in pressure as it does so. The section of the casing rings, as seen in the top cover in Fig. 282, clearly shows the gradually increasing space provided for the steam to expand in volume as it passes through the turbine. The machine is of the reaction type.

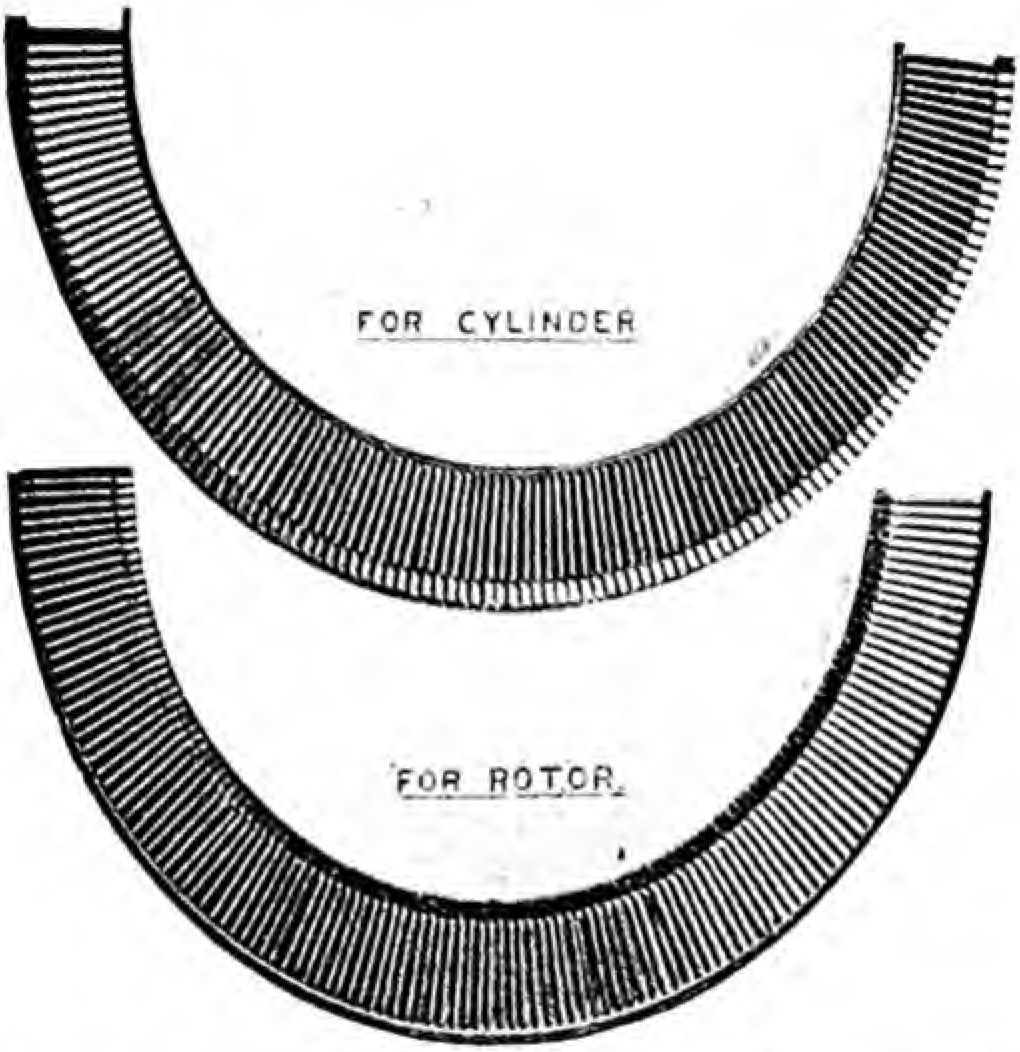


FIG. 283.—Half rings of blades in the Willans & Robinson and Sankey system.

Each pair of fixed and rotating rings may be called a stage. In each stage the steam undergoes a small drop in pressure and increase in volume, and a portion of its energy is abstracted from it by the rotating ring and so given to the shaft. It will be noticed that twice in the length of the turbine the rings have a sudden change in diameter. This arrangement is made in order that sufficient volume may be provided for the expanding steam. The same result could be obtained with rings all having the same diameter, but in this case the blades would be either too long radially at the low pressure end, or too short at the high pressure

end. The method adopted provides for expansion and secures a practical length of blade throughout.

Blading.—The system of blading illustrated is covered by the Willans & Robinson and Sankey patents, and is very effective. The blades for both casing and rotor are assembled usually in half

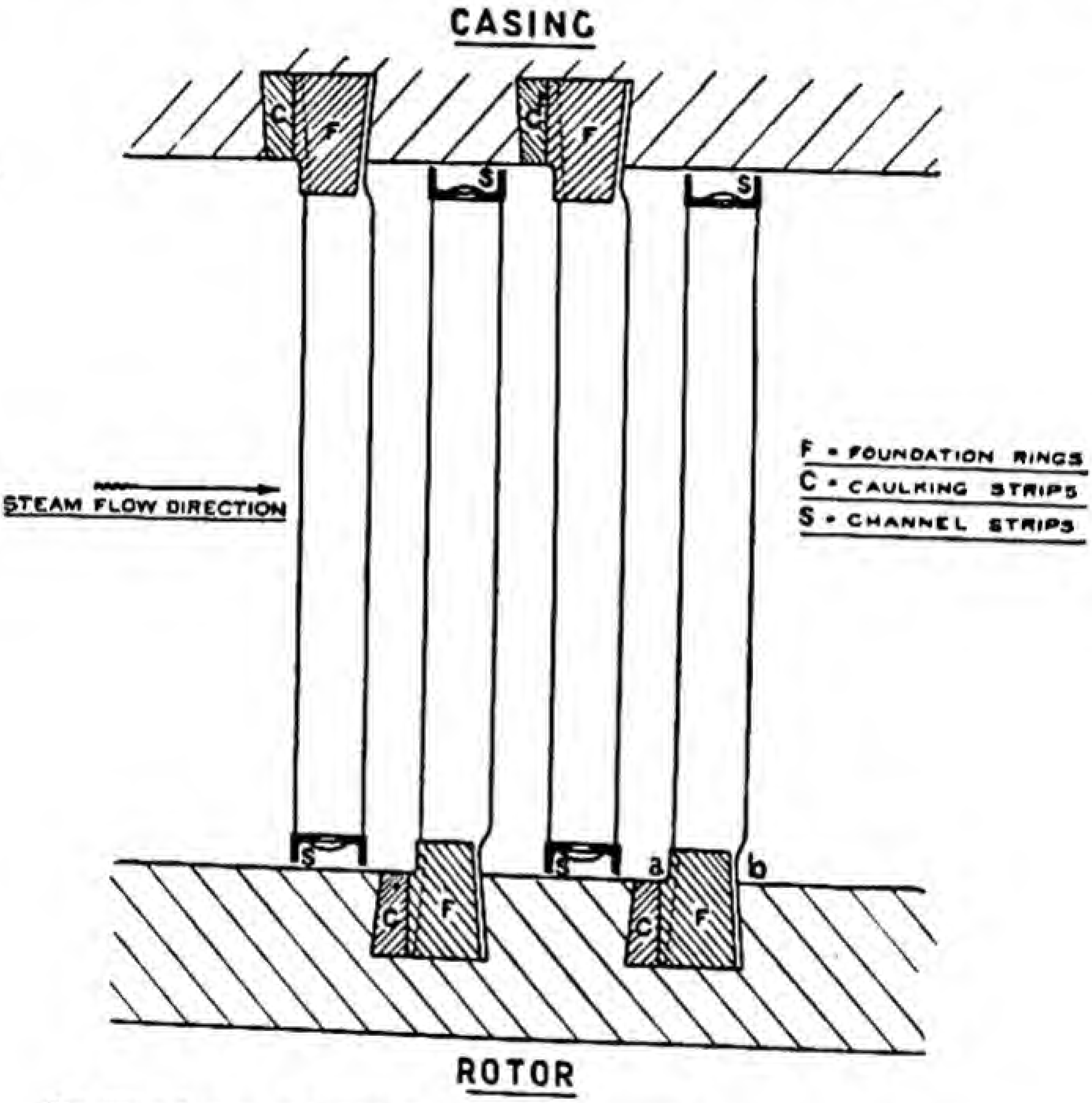


FIG. 284.—Section of the blading system covered by the Willans & Robinson and Sankey patents.

rings (Fig. 283), or in smaller segments if the turbine is very large. These segments are subsequently secured by caulking strips in dovetail grooves in the shaft and casing. In Fig. 284, F is a foundation ring which is cut with dovetail slots to receive the roots of the blades, as shown by the dotted line. A section through ab is given separately in Fig. 285, showing the slots in the foundation ring and also the shape of the roots of the blades. The blades are pressed or stamped to the correct section where

the steam passes over them, the roots or tangs being also pressed or stamped to the proper dovetail shape. The other ends of the blades are stamped with a small tongue for riveting into the channel section shrouding *S*. Both foundation ring and shrouding are slotted and punched on automatic dividing machines, so that both the pitch and angle of the blades are determined accurately.

The blades are assembled to the proper number into a complete half ring by lightly riveting the tongue of each blade into the channel shrouding, and pushing the wider part of the tang of the blade into one of the slots in the foundation ring. The light riveting permits a certain degree of freedom of the blades in the shrouding, so that they may be entered easily into their places in the foundation ring. The half ring is then placed in the groove provided for it in the shaft or casing, and is fixed by means of the caulking strip *C*, which gives strength and rigidity to the whole. The final riveting of the blades into the channel is completed after the ring is secured in the grooves.

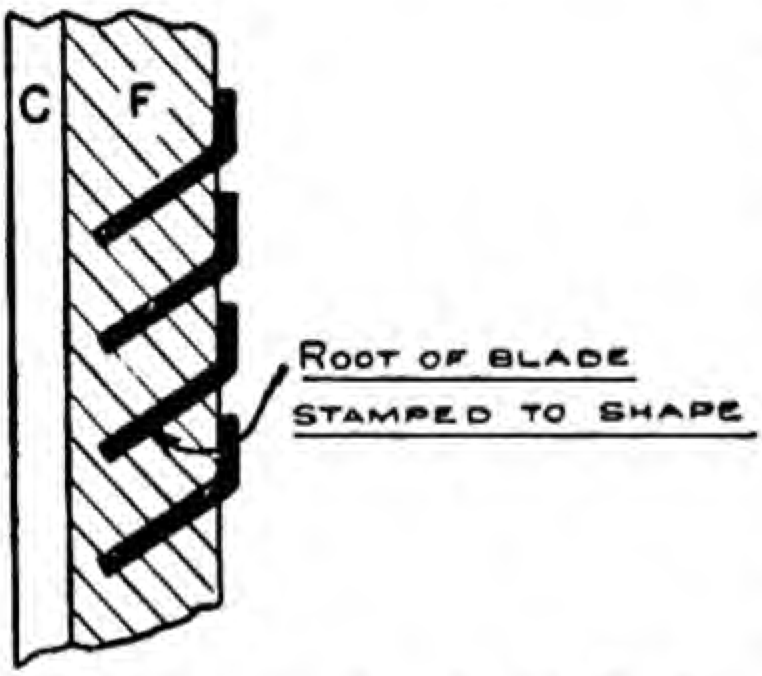


FIG. 285.—Section through *a b* in Fig. 284.

It will be observed from Fig. 284 that the groove is dovetailed or under-cut on both sides. The foundation ring *F* has one side at the same angle as the side of the groove, the smaller portion of the root of the blade coming between this side of the ring and the side of the groove, this portion of the blade is thus nipped, after *C* has been caulked into place, between the foundation ring and the side of the groove. Thus there are two very effective preventives to the blade becoming loose under the action of centrifugal force. The rotor, in which the grooves to receive the rotating rings are cut, is a hollow steel forging, turned inside and outside to insure perfect balance. The end shafts are shrunk into the forging and are also bolted against radial faces on it, thus securing rigidity of structure unlikely to be affected by unequal expansion and the heavy working stresses.

The channel shrouding serves to stiffen and protect the blades, and is also instrumental in preventing the steam from leaking past over the tips of the blades instead of flowing through. This utility is due to the **eddy**, which will appear in any steam leaking over the first edge into the enlarged section of the channel. It is well known that the resistance offered to eddy flow is much higher than that of steady flow—hence the utility of the channel in minimising leakage. It will also be understood that, owing to the large number of stages, the pressure drop from one ring to the next succeeding is small. There is thus only a small pressure difference on the two sides of any one ring and consequently only a small force impelling leakage.

Balancing the end thrust.—As the steam flows axially through the turbine, it will exert a resultant force tending to carry the rotor in the same axial direction. This is balanced by causing the steam to react on balance pistons or collars attached to the rotor. Two of these may be observed on the right of the inlet ring of blades in Fig. 282.

Governing is effected by means of a powerful centrifugal governor driven by worm gearing from the turbine shaft. This governor is connected by means of a single lever to the main regulating valve controlling the steam supply. Friction is almost eliminated by the use of ball bearings to all working parts. A secondary governor is arranged to shut the turbine down should the speed exceed a prearranged limit.

Oil is supplied to all the bearings by means of a rotary pump driven direct from the turbine. Thus, immediately the turbine starts, lubrication begins. The oil returns from the bearings to a tank in the steam end base plate (Fig. 281), where it is cooled by water circulation and redelivered to the bearings by the pump. The bearings can be flushed with oil before starting by means of a hand pump. As will be observed from Fig. 282, the machine can be opened for inspection by folding back the hinged covers without in any way interfering with the working parts.

Complete plant.—Fig. 286 shows the general arrangement of a 1400 kilowatt Willans-Siemens Direct Current Turbine set at Glasgow. The electric generators are direct coupled to the turbine shaft. The surface condenser *A*, motor-driven air pumps

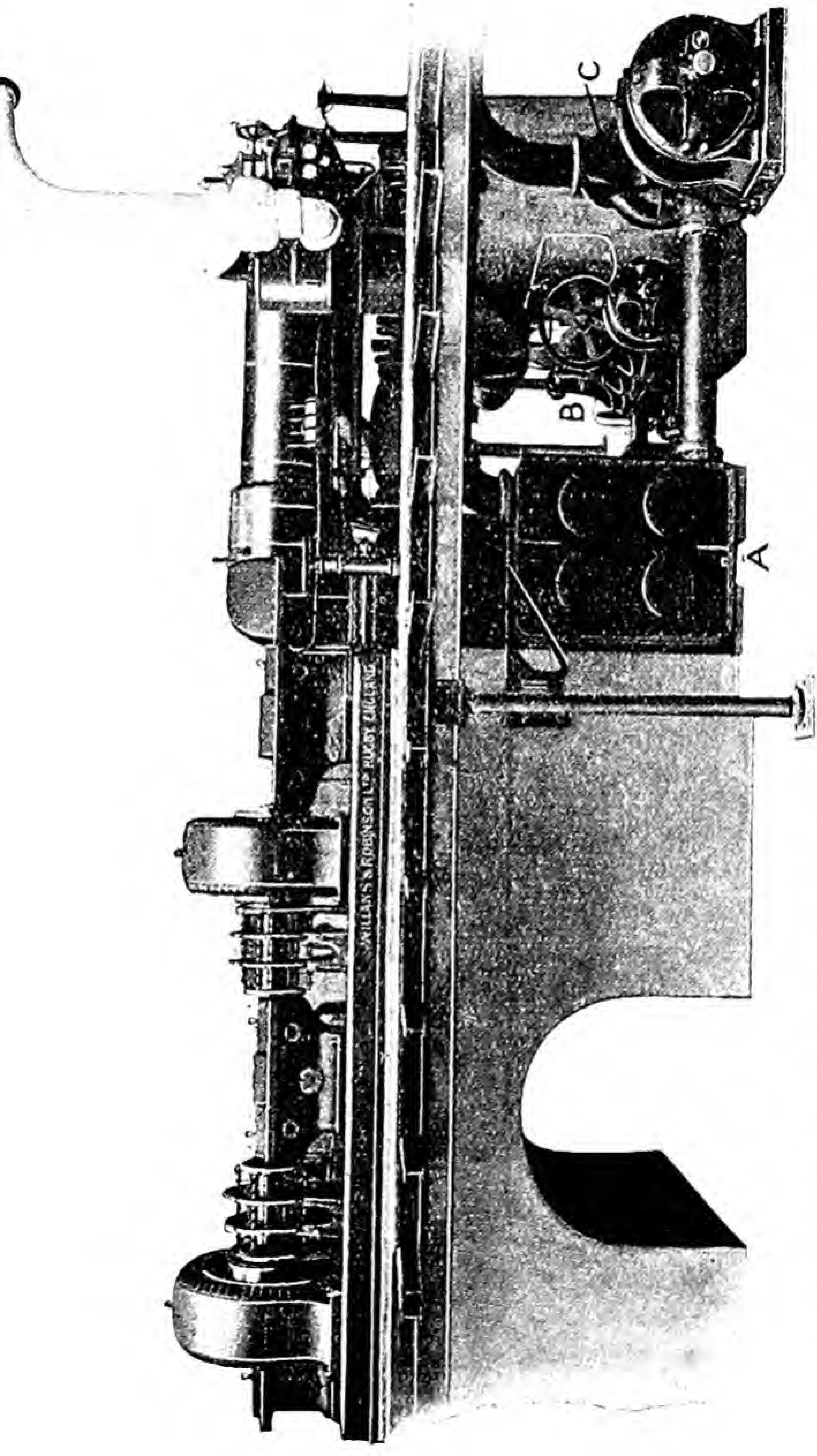


FIG. 286.—Arrangement of 1400 kilowatt Willans-Siemens Direct Current Turbine set at Glasgow.

B, and the motor-driven centrifugal circulating water pump *C*, are situated in the basement below the turbine and generators.

Westinghouse double-flow turbine.—This turbine is constructed by Messrs. The British Westinghouse Electric & Manufacturing Co., Ltd., to whom the author is indebted for the illustration. In it is found a combination of the Curtis and the Parsons turbines. The steam is admitted to a steam belt *A* (Fig. 287), and then flows axially in both directions through the turbine wheels. Leaving *A*, the steam is first expanded through nozzles *B*, in which its pressure falls and the velocity correspondingly increases. The nozzles direct the steam currents through the buckets *C*, *C* fixed to the rotating wheel, when the direction is again changed by the fixed row of blades *D*, *D* before passing the second ring of moving buckets *E*, *E*. This constitutes the Curtis portion of the turbine. In this portion about one-third of the energy available in the steam is absorbed. The steam is then expanded through the Parsons blades *F*, fixed to the rotating spindle and to the cylinder, gradually dropping in pressure as it does so and imparting energy to the turbine shaft, and is discharged finally to the exhaust openings *H*, *H* at each end of the turbine cylinder. As both halves of the turbine on either side of the steam belt are identical, the steam pressures balance and there is no end thrust on the shaft.

To seal the inside of the turbine, and so prevent leakage of air inwards, water glands are used at *J*, *J*, consisting of small paddle wheels, somewhat similar to the wheel of a centrifugal pump, so designed as to back up water pressure in opposition to the pressure of the atmosphere. The main bearings *K*, *K*, are of cast-iron, lined with white metal, and are housed in spherical seatings to allow of any slight defect in alignment. To maintain the spindle and attached rings of buckets in their correct position in relation to the fixed cylinder buckets and nozzles, a thrust block *L* is provided. This is made in halves, the bottom half being fitted with collars bearing against the collars of the spindle on one side, and the top half having its collars bearing against the spindle collars on the other side.

An emergency governor, not shown, is fitted on the main shafting at *M*, designed to come into play at a speed of some 10 per cent. above the normal working speed of the turbine, automatically

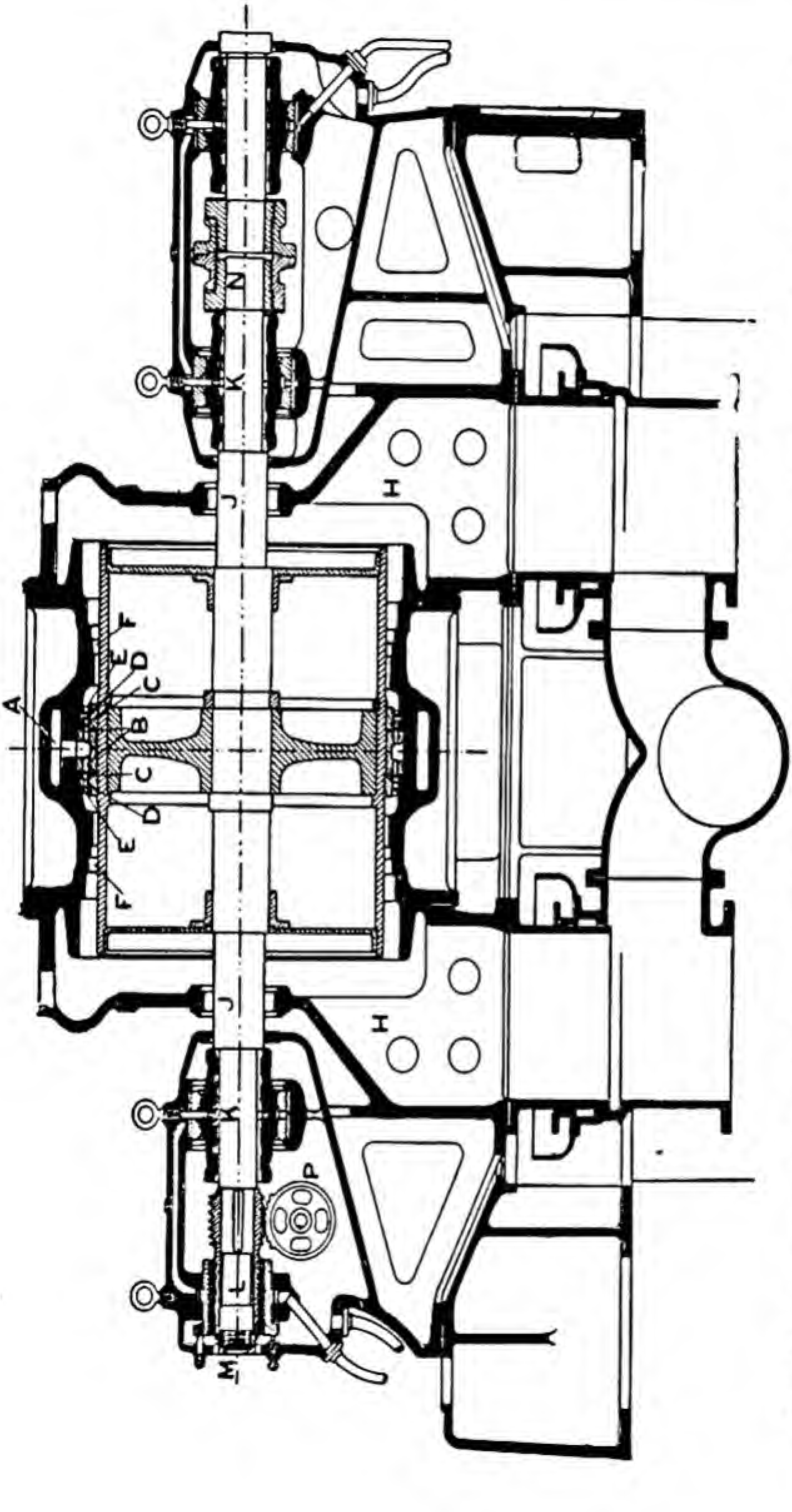


Fig. 287.—Section of the Westinghouse double-flow steam turbine, combining the Curtis and the Parsons systems.

shutting it down. The main governor controls the admission of steam to the steam belt by throttling. The governor gear is driven through a worm wheel P from the worm fixed to the turbine spindle.

Forced lubrication is used, oil being delivered to each bearing from a main trunk pipe under a few pounds pressure. The oil is returned from the bearings to an oil tank to be forced again by the oil pump through an oil cooler to the bearings.

Exhaust steam turbines.—With reciprocating engines it is not possible economically to carry the expansion of the steam beyond a certain point owing to the enormous size of low pressure cylinder required. The heat and mechanical losses would quite outbalance any theoretical gain which might otherwise be obtained. With the turbine it is quite convenient to have high ratios of expansion, as the dimensions of the parts are moderate. Thus the steam turbine can utilise economically steam at a much lower pressure than a reciprocating engine. Hence the importance of attention to the design of the condensing arrangements of a turbine plant so as to secure a high vacuum.

An interesting application of this principle has been made in cases where non-condensing reciprocating engines are already at work. The exhaust steam from the engines is collected, instead of allowing it to discharge to waste into the atmosphere, and is delivered to turbines specially designed to work with low pressure steam. The general efficiency of the whole plant may thus be increased largely. In some cases even double the power may be obtained from the modified plant without any increase in the original steam consumption.

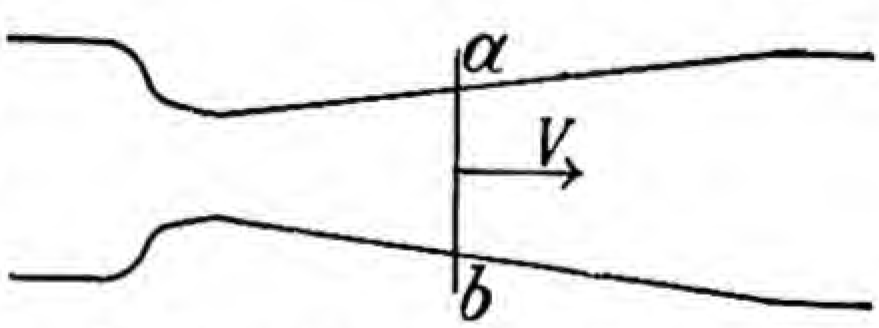


FIG. 288.—Flow of a fluid through a nozzle.

Flow of a fluid through a nozzle.—The consideration of the flow of steam through nozzles is of importance in connection with the subject of turbines. In the case of a liquid the problem is simple as the property of expansibility is absent. Consider a liquid such as water flowing through a nozzle as illustrated in Fig. 288. Let A be the cross-sectional area in square feet of any

section such as ab and V the velocity of the water when passing through this section. It being assumed that there is continuity of flow, *i.e.* no spaces unfilled with water, precisely the same volume of water will pass any section of the nozzle per second. The weight of water passing ab per second will be equal to ρVA lbs. where ρ is the weight in lbs. of a cubic foot, and this will be constant for all sections, hence

$$\rho VA = c,$$

c being constant, and therefore

$$V = \frac{c}{\rho A},$$

or the velocity at any section varies inversely as the area of the section, since both c and ρ are constant.

Flow of steam through a nozzle.—Assuming the same condition of continuity of flow, and taking account of the change in density of the steam, due to its expansion as it passes through the nozzle,

Let V = the velocity in feet per sec. at any section ab (Fig. 288),
A = the area of ab in square feet,
ρ = average weight of steam in lbs. per cubic foot under the conditions of pressure, etc., at ab,
W = weight of steam in lbs. passing ab per sec.;

then

Volume passing $ab = AV$ cubic feet per second,
Weight " " $= \rho AV$ lbs. per second.

As expansion goes on in the steam flowing through the nozzle, the pressure falls and the velocity increases; the volume per lb. weight increases, consequently the weight of a cubic foot will be lower. It is found that the product of ρ and V at first increases and then decreases. The maximum product ρV occurs at the section of the nozzle where the pressure bears a ratio of about 0·58 to the initial pressure. Up to this section the product ρV increases, hence A is made to diminish by making this portion of the nozzle **convergent.** Supposing the nozzle to end at this section and the steam to be discharged direct from it into the exhaust chamber, the pressure in the exhaust chamber may be about 0·58 of the initial pressure, *i.e.* the same as that of the terminal section of the nozzle. Should the pressure in the exhaust chamber be less than this value, the pressure in the terminal section of the nozzle

will be the same as before, but expansion of the steam will be completed by lateral spreading after discharge from the nozzle. To secure further expansion in the nozzle to a pressure lower than 0·58 of the initial pressure, the nozzle should be extended beyond the above-mentioned section. As the product ρV is now diminishing, the area of the sections must be increasing, hence the nozzle diverges in the portion beyond the section where ρV attains its maximum value. This portion is made conical usually, the angle of the cone being about 10°. The cone must not diverge so rapidly that there is risk of the steam becoming separated from the sides of the nozzle.

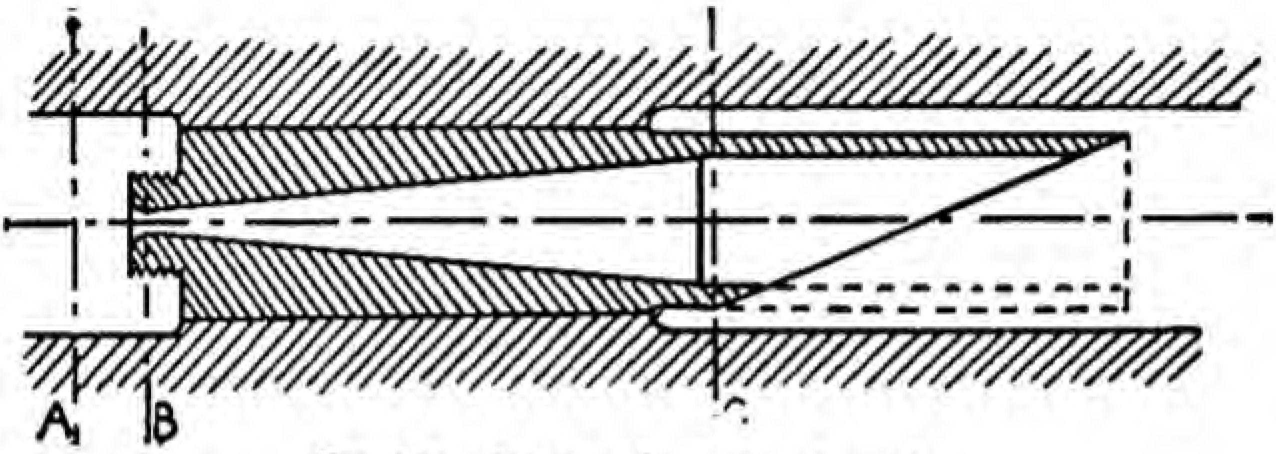


FIG. 289.—Section of a de Laval nozzle.

For example, take a nozzle in which it is intended to expand completely steam at 200 lbs. per square inch to a terminal pressure equal to that in the discharge chamber, which is maintained at 28″ vacuum. Assuming that the admission steam is dry, the pressures, etc., in the different sections will be as follows (Fig. 289):

Section A:

Pressure: 200 lbs. per square inch above atmospheric.
Percentage of moisture in the steam: 0.

Section B (the smallest section of the nozzle):

Pressure: 110 lbs. per square inch above atmospheric.
Percentage of moisture in the steam: 4.
Velocity of the steam: 1500 feet per sec.
Specific volume of the steam: 3·5 cubic feet per lb.

Section C (the largest section of the nozzle):

Pressure (28″ of vacuum): 2″ of mercury absolute pressure.
Percentage of moisture in the steam: 24.
Velocity of the steam: 4127 feet per second.
Specific volume of the steam: 256·8 cubic feet per lb.

The ratio of the areas of the largest and smallest sections of this nozzle would be 27·234. The ratio of the corresponding diameters would be 5·219. Thus, supposing the diameter at B to be $\frac{1}{4}''$ (or 6 mm.), the diameter at C would be nearly $1\frac{1}{4}''$ (or 31·31 mm.). This nozzle will maintain a constant discharge of 479 lbs. per hour of steam initially dry and saturated, neither more nor less.[1]

Angles of blades and nozzles.—In designing a turbine, the mechanical principles depended upon are: (*a*) The steam must enter the blade of the revolving wheel without **shock**; (*b*) the energy of the steam leaving the turbine wheel must be as low as possible, attained by causing the leaving velocity to have the minimum practical value.

It is shown in mechanics that if two bodies come into collision, the impact always causes a loss of energy—hence the necessity of avoiding anything like collision or shock when the particles of steam come into contact with the revolving blades. Shock is avoided by so relating the angle of the nozzle and the angle which the entering edge of the blade presents to the nozzle that the steam slides on to the blade without impact. These angles are related to the velocity of the entering steam and to the velocity of that part of the revolving wheel where the blade is attached.

EXAMPLE. Let the velocity of the steam entering an impulse turbine wheel be 4000 feet per sec. and the angle to the plane of the wheel at which it is directed by the nozzle 20°. Let the velocity of the wheel blade be 1380 feet per second. Draw *ab* (Fig. 290), making 20° with the plane of rotation *ac*, and make it to scale to represent $v_1 = 4000$ feet per second. Let *ac* to the same scale represent $V = 1380$ feet per second. Complete the parallelogram of velocities *acbd*. Then *ad* will represent v_r, the velocity of the entering steam relative to the blade. To enable the steam to enter the blade without shock, the direction of the blade section at *a* should be along *ad*; the steam will then slide on to the blade without impact. Measuring v_r to scale, it is found to be 2720 feet per second, and to be at an angle of 30° to the plane of rotation of the wheel. The entering edge of the wheel must therefore be at 30° to the plane of rotation.

The blade may be curved to the arc of a circle as shown. Assuming that the leaving angle of the blade at *e* is also 30°, we find the absolute

[1] See lecture delivered at the Yorkshire College on the de Laval steam turbine by Mr. Konrad Anderson.

leaving velocity of the steam thus. Assuming frictionless flow of steam along the blade, it will be discharged from e with a velocity made up of two velocities, viz. (*a*) the velocity of 2720 feet per second, causing it to slide along the blade and represented in Fig. 290 by ef; (*b*) the velocity of 1380 feet per second which it has in virtue of being in contact with the blade, represented by eg. Find the resultant of

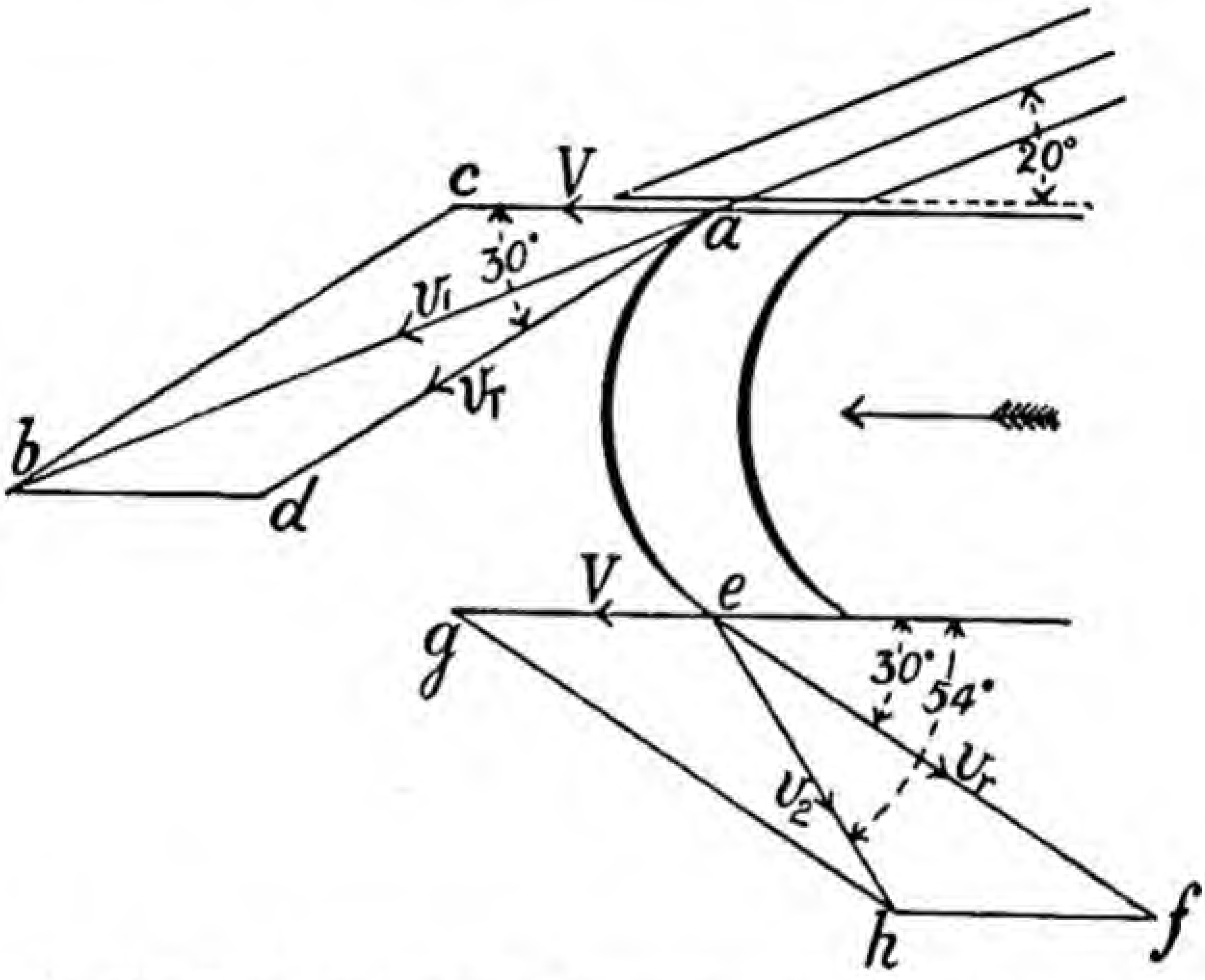


FIG. 290.—Flow of steam through the nozzle and past the buckets of an impulse turbine.

these by the parallelogram of velocities $efhg$, giving the final leaving velocity v_2 represented by $eh = 1680$ feet per second, and making an angle of 54° with the plane of the wheel.

Work done by the steam.—We may now calculate the work done by the steam on the wheel.

Let W = weight of steam in lbs. supplied per sec.,

v_1 = initial velocity of steam in feet per sec.,

v_2 = leaving velocity of steam in feet per sec.,

Then

Kinetic energy available in the entering steam

$$= \frac{Wv_1^2}{2g} \text{ ft.-lbs. per sec.}$$

Kinetic energy carried away per sec. by the leaving steam

$$=\frac{Wv_2^2}{2g}\text{ ft.-lbs.,}$$

Energy which can be converted per sec. into mechanical work

$$=\left(\frac{Wv_1^2}{2g}-\frac{Wv_2^2}{2g}\right)\text{ ft.-lbs.}$$

$$=\frac{W}{2g}(v_1^2-v_2^2)\text{ ft.-lbs.,}$$

$$\text{Horse-power available}=\frac{W}{2g}(v_1^2-v_2^2)/550.$$

We may also write:

Energy which can be converted per sec. into mechanical work per lb. of steam supplied

$$=\frac{v_1^2}{2g}-\frac{v_2^2}{2g}$$

$$=\frac{v_1^2-v_2^2}{2g}\text{ ft.-lbs.}$$

Initial energy supplied per lb. of steam per sec.

$$=\frac{v_1^2}{2g}\text{ ft.-lbs.}$$

Hence,

$$\text{Efficiency}=\frac{\text{energy derived}}{\text{energy supplied}}$$

$$=\frac{\dfrac{v_1^2-v_2^2}{2g}}{\dfrac{v_1^2}{2g}}$$

$$=\frac{v_1^2-v_2^2}{v_1^2}$$

$$=1-\frac{v_2^2}{v_1^2}.$$

Applying these results to the velocities obtained in the above example:

$$v_1=4000\text{ feet per sec.}$$

$$v_2=1680\text{ feet per sec.}$$

$$\text{Efficiency} = 1 - \frac{v_2^2}{v_1^2}$$
$$= 1 - \left(\frac{1680}{4000}\right)^2$$
$$= 0.8235$$
$$= \underline{82.35} \text{ per cent.}$$

It may be noted that, to secure the highest possible efficiency, the velocity of the wheel blades should be about 47 per cent. of that of the entering steam. With a steam velocity of 4000 feet per second this would give a wheel velocity of 1800 feet per second or about 21 miles per minute. This speed is impracticable for various mechanical reasons, hence the speed at the centre of the buckets does not exceed in practice 1380 feet per second, giving a velocity of wheel rim of about 16 miles per minute.

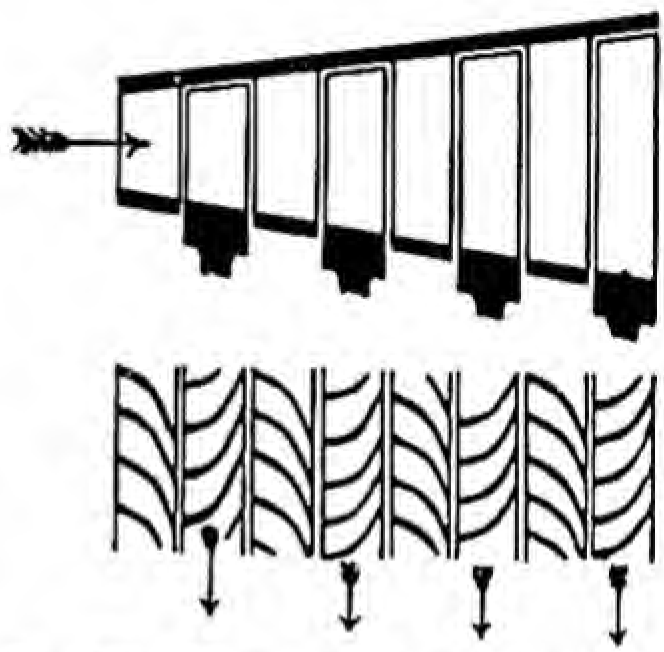


FIG. 291.—Diagrammatic representation of a reaction steam turbine.

The reaction turbine.—The reaction turbine may be described as a number of rotating bladed wheels arranged to cut at right angles a diverging nozzle through which steam is flowing and undergoing expansion (Fig. 291). The blades of the wheel pass through the nozzle; fixed blades in the nozzle between each pair of rows of moving blades direct the steam properly. The idea is to limit the drop in pressure in any one stage, consisting of one set of fixed and one set of moving blades, and thus to limit the velocity of the steam which has to be received by any one set of rotating blades. The speed of the whole machine may thus be made quite moderate compared with the enormous velocities attained in the single stage turbine. It will, of course, be understood that the nozzle in practice is carried right round the turbine casing so as to embrace the whole of the moving blades simultaneously (Fig. 282).

In Fig. 292, A is a single guide plate receiving steam with a velocity v_0. Expansion as the steam passes the guide blade causes

this to increase to v_1, while the direction is changed from that of v_0 to ab. The moving blade B receives the steam at a. To obtain the entrance angle of the blade, proceed as before, V being the velocity of the moving blade and v_r the velocity of the entering steam relative to the blade. The blade at a will be tangential to ad. In passing over the moving blade from a to e the steam will expand further, causing v_r to increase to v_R at the leaving edge e. The absolute velocity v_2 of the leaving steam is found by compounding v_R with V.

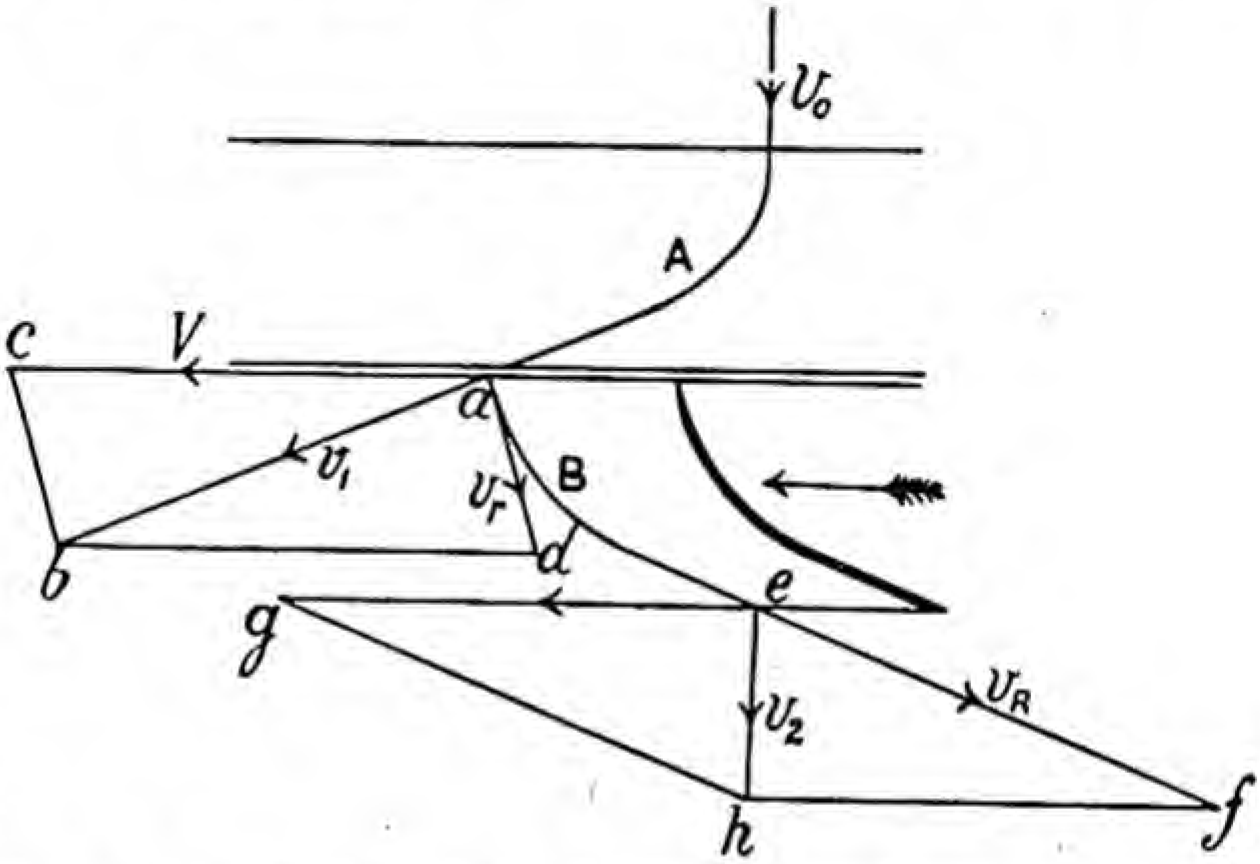


FIG. 292.—Flow of steam past the guide blades and moving blades of a reaction turbine.

Sources of waste in turbines.—Professor Rateau separates the sources of loss in turbines as follows:[1]

1. **Internal losses**, due to the friction and eddying of the steam passing over the fixed and moving blades.

2. **External losses**, due to (a) leakage of steam past the clearance spaces; (b) the friction of the revolving wheels upon the surrounding steam; (c) the friction of the bearings.

For a turbine of his own type (multicellular), of 1500 B.H.P., having a speed of 1500 revolutions per minute, he finds waste due to internal losses amounting to 31 per cent.; due to leakage

[1] *Proc. Inst. Mech. Eng.*, June, 1904.

and bearing friction 1·5 per cent.; due to friction of the wheels upon the steam 2·5 per cent., making a total external loss of 4 per cent. The net efficiency will then be

$$\text{Efficiency} = 0{\cdot}69 \times 0{\cdot}96 = 0{\cdot}66$$
$$= 66 \text{ per cent.}$$

This result may be used for calculating the steam consumption of this turbine under given conditions of pressure and superheat.

It should be noted that the complicated action of cylinder walls in producing initial condensation and re-evaporation, as in the reciprocating engine, is entirely absent in turbines. When the turbine has settled down to its work, the temperature of vanes, casing, etc., at any given point in the machine remain practically steady. A further advantage lies in the absence of reciprocating parts, which enables practically perfect balance to be obtained.

EXERCISES ON CHAPTER XVII.

1. Distinguish clearly between impulse and reaction steam turbines.

2. Explain the action of the de Laval turbine. Make a sketch showing the wheel and one of the nozzles. Why is it necessary to use a high velocity of rotation in this type?

3. Sketch and describe the nozzles, fixed blades and revolving wheels of a four stage Curtis turbine.

4. Explain clearly the reasons for the speed of the Curtis turbine being much lower than that of the de Laval, both being impulse turbines.

5. Sketch and describe the fixed blades and revolving wheels of a Parsons turbine.

6. Show clearly by sketches and describe the method of fixing the blades in any steam turbine. What precautions have to be taken?

7. Explain and give sketches showing the construction of any steam turbine governor.

8. Explain how end thrust is guarded against in any steam turbine.

9. Explain why a larger ratio of expansion can be used economically in a steam turbine than in a reciprocating engine.

10. Explain in general terms the flow of steam through a diverging nozzle.

11. Make a sketch with the angles approximately correct of the nozzle and blades of an impulse turbine. What are the conditions to be satisfied?

12. Answer question 11 for a reaction turbine.

13. Give a brief summary of the sources of wasted energy in steam turbines.

CHAPTER XVIII.

ENGINE AND BOILER TRIALS.

Experimental steam plant.—The nature and scope of the experimental work which the student will perform will, of course, depend largely on the equipment as regards engines and boilers which is available. The following general description of the plant at the West Ham Technical Institute may be of interest. Some of the easier experimental work which can be carried out by students in the elementary classes is included, and will be helpful to the student no matter what may be the equipment of his own laboratory. It may be noted here that each experiment on an engine or boiler should form one of a series in which the conditions of running are being modified for each experiment. No single student would probably be assisting in the whole of any given series, but if the experimental results up to date are made available to him, he is likely to take a more intelligent interest in that part at which he is working. Isolated experiments are of little value and are apt to degenerate into mere playing about with costly machinery.

Experimental steam engines.—To gain any definite information regarding the working of a steam engine, the engine should be so designed that the conditions may be altered, one at a time. This means that an engine specially constructed for experimental work should, if possible, be available.

In Figs. 293 and 294 is shown such a set of engines, constructed by Messrs. E. T. Hindley & Sons. The set comprises a pair of vertical engines with inverted cylinders and coupled crank shafts. Each engine is complete in itself, and the engines can be disconnected by drawing the coupling bolts and taking out a loose

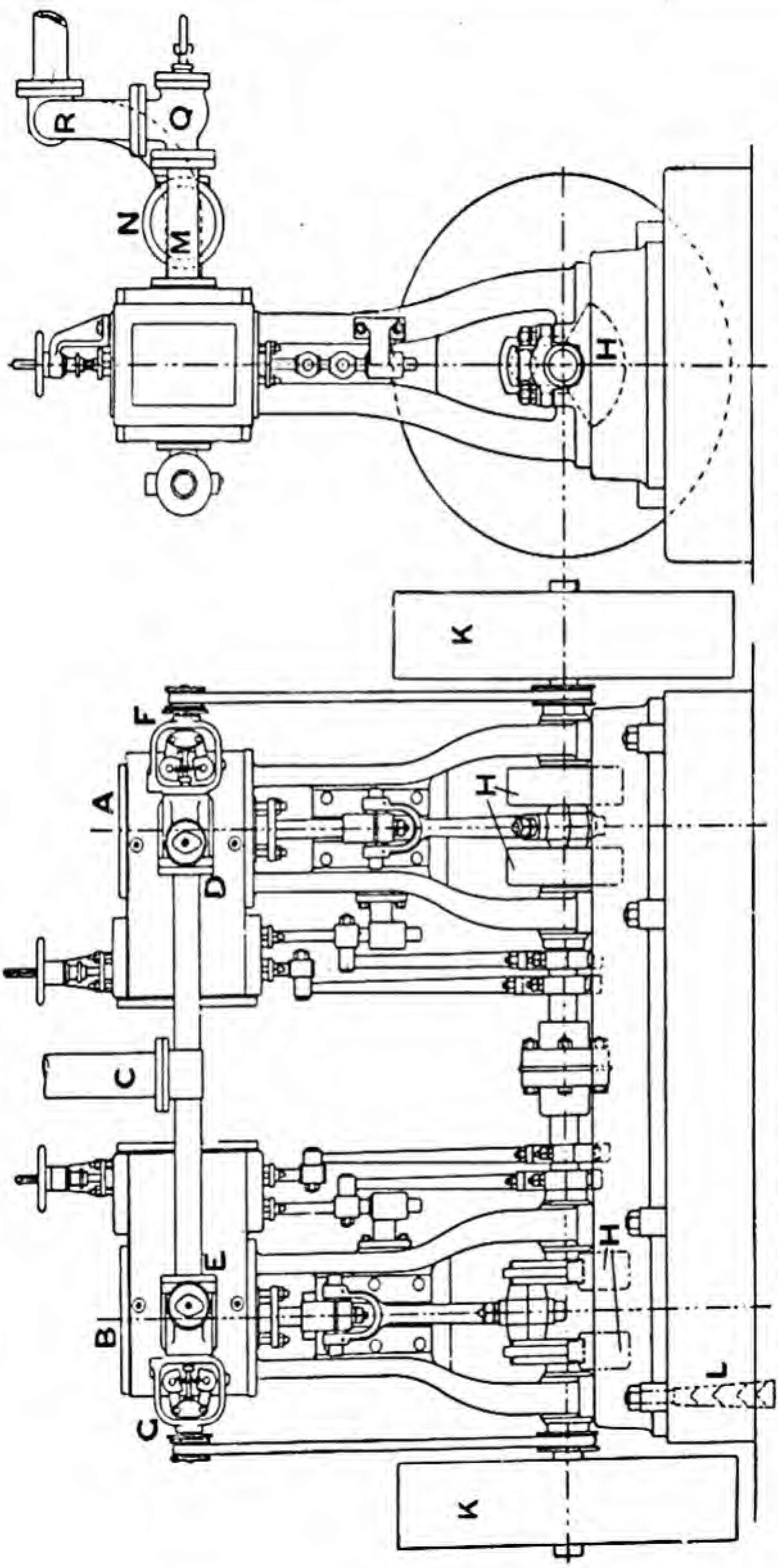


FIG. 293.—Front and end elevations of the experimental steam engines at the West Ham Technical Institute.

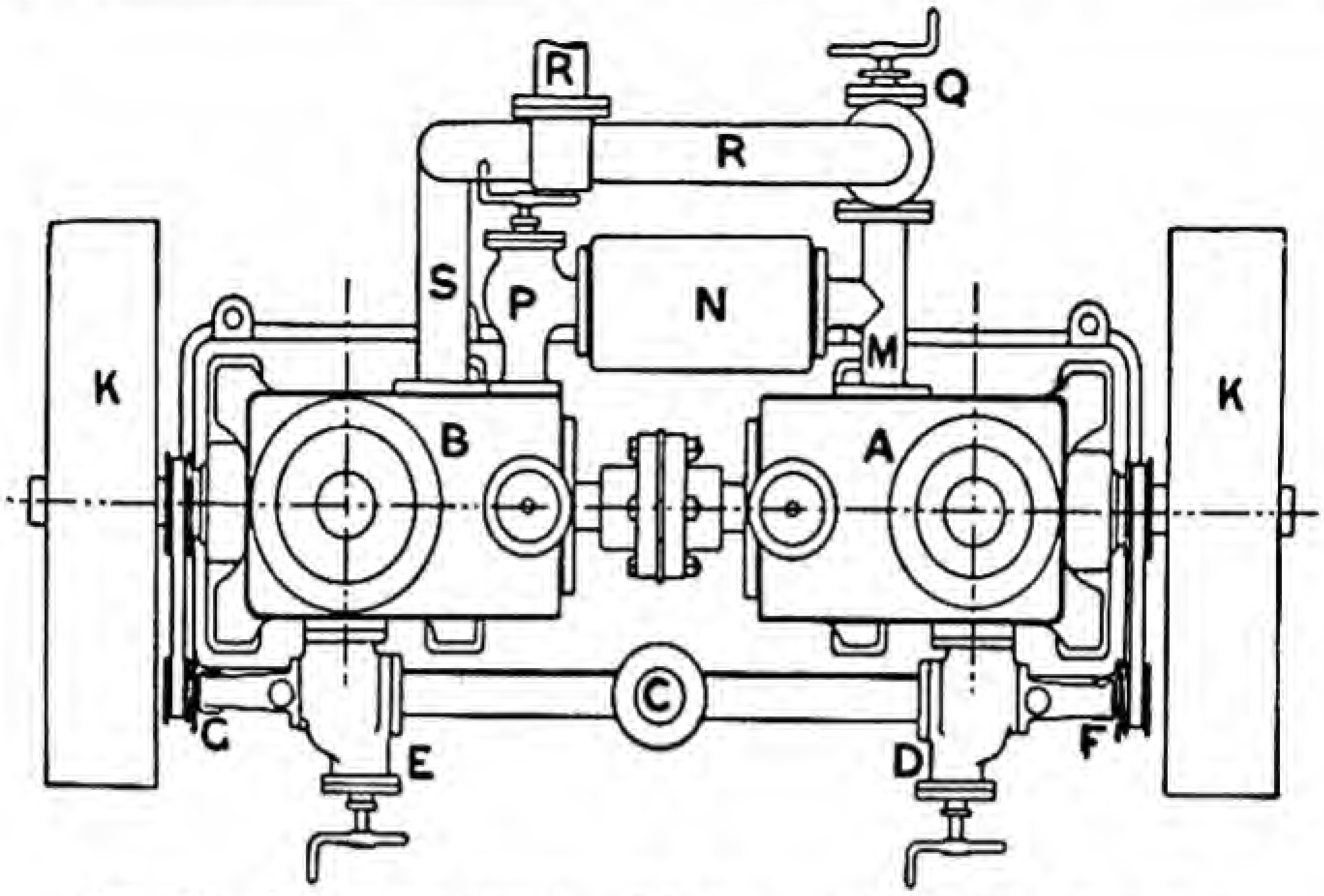


FIG. 294.—Plan of the experimental steam engines at the West Ham Technical Institute.

plate fitted between the faces of the coupling. The crank shafts will then run independently without the couplings touching one another. The crank shafts may be coupled so that the cranks are together, or at 90°, 180°, or 270°.

Both engines are alike in all respects excepting their cylinders. In Figs. 293 and 294, *A* is a cylinder 6″ in diameter × 8″ stroke; *B* is a cylinder 10″ in diameter × 8″ stroke. *A* is jacketed, *B* is unjacketed.

Steam is brought to the engine through a steam pipe *C*, a separator being placed in the pipe just above the engine. The pipe *C* divides right and left, leading to stop valves *D* and *E*, by means of which steam may be turned on to either cylinder. Centrifugal governors *F* and *G* control the speed.

Cylinder details.—The 6″ cylinder is shown in detail in Fig. 295. Steam enters the steam chest through the passage *A*, and is distributed to the cylinder by a Meyer's valve gear. The cut-off may be set to occur at any point between 0·25 and 0·8 stroke by rotating the hand wheel *B*. This wheel is capable of rotation but not of axial movement, being restrained by collars on each side of the bracket *C*, and, when rotated, produces a similar movement in the valve spindle *D*. To enable this to take place, a feather is secured to the wheel and works in a keyway in the valve spindle

as the latter reciprocates. The valve spindle has right and left handed square screws cut on it to receive the gun-metal nuts *E*, *E*, which operate the expansion valves. To show the point of cut-off

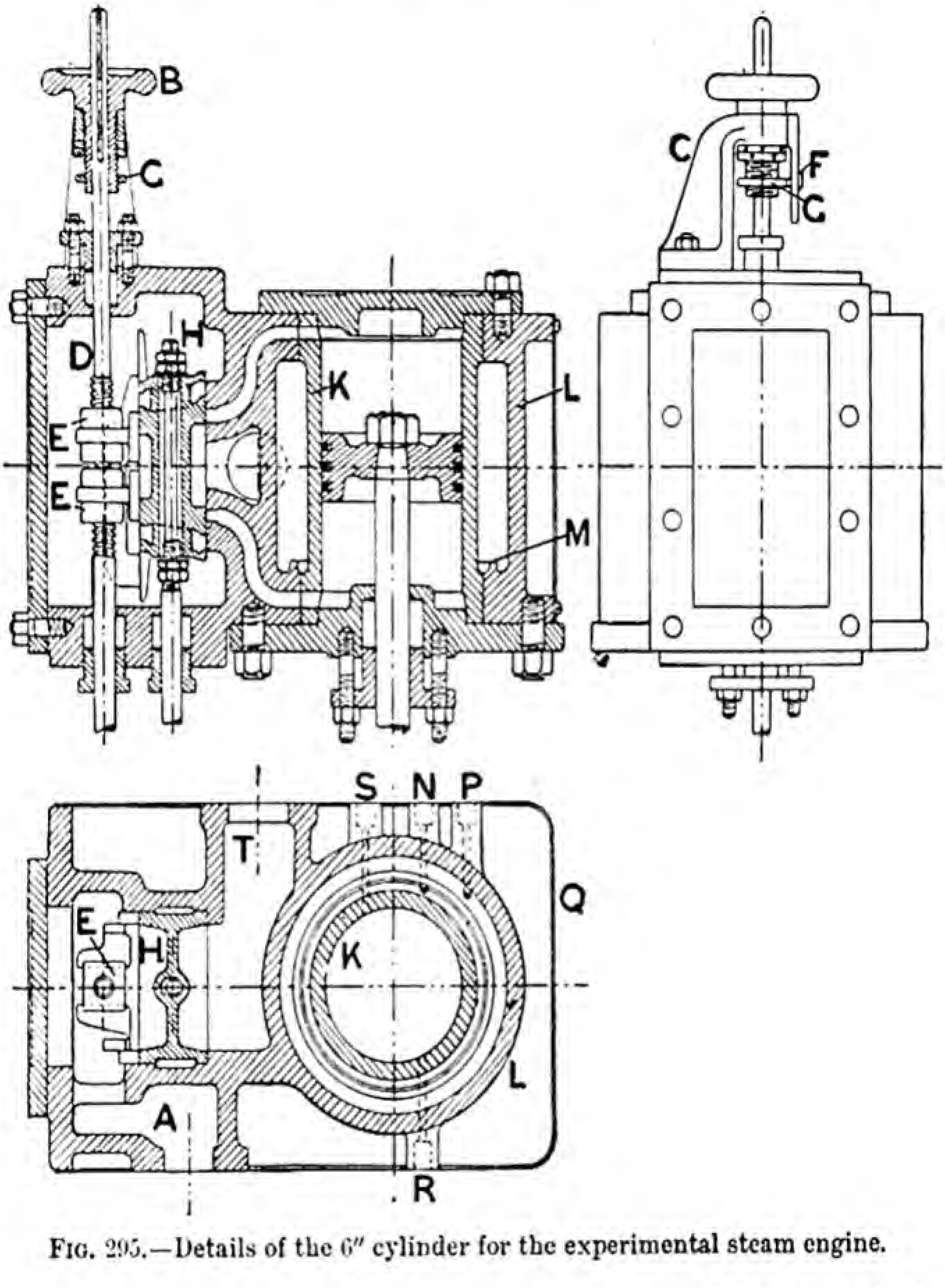


FIG. 295.—Details of the 6″ cylinder for the experimental steam engine.

for any setting of the expansion valves, a pointer *F* is attached to a screwed collar *G*, which fits a screwed extension of the boss of the hand wheel. The collar is held from rotation, and consequently gives axial movement to the pointer when the hand wheel is rotated. The eccentric driving the expansion valve is set at 180° to the crank.

The main valve *H* has ports through it, as already described (p. 122). This valve is set permanently to cut off at 0·8 stroke, thus determining the latest possible cut-off of the gear.

The cylinder is fitted with a liner *K*, forced home and held in place by the top cylinder cover. The space between the liner and the barrel *L* forms the steam jacket. In larger cylinders, the covers may be jacketed also, but troublesome construction would be introduced in carrying this out in the small cylinder under consideration. There is a ridge *M* at the bottom of the jacket, forming two gutters running right round. The inner gutter is drained through *N* and the outer through *P*. It is thus possible to separate the water of condensation coming from the outer wall of the liner from that coming from the inner wall of the barrel. The cylinder is lagged with non-conducting material held in place by steel sheeting *Q*. Indicator cocks are attached at *R*, and drain cocks and relief valves at *S*. During exhaust the steam is discharged through *T*, either to the condenser or to the 10″ cylinder as may be desired.

The 10″ cylinder is identical with the 6″ cylinder excepting that it has no liner. The barrel *L* forms the cylinder walls.

Further details.—Referring again to Fig. 293, it will be noticed that each engine is balanced by means of heavy masses, *H*, attached to the cranks on the side opposite the crank pins. Rope brakes for ascertaining the B.H.P. are passed round the flywheels *K*, *K*; the rims of these wheels are channelled to receive cooling water (p. 105). The engine is secured to its concrete foundations by means of ragged bolts, one of which is indicated at *L*.

The exhaust pipe arrangement is shown in the plan and end elevation. The exhaust steam from the 6″ cylinder *A* enters the pipe *M* and may be directed into a receiver *N*, and so through a valve *P* into the 10″ cylinder steam chest. Otherwise the steam may be directed through a valve *Q* into a pipe *R* leading to the condenser. In compound working, the valve *P* is open and *Q* is closed; steam then flows from *A* through the receiver and 10″ cylinder, and is exhausted to the condenser from this cylinder through *S* and *R*.

If the 6″ engine alone is running, the valve *P* is closed and *Q* opened, when the exhaust steam will enter the condenser through the pipes *M* and *R*. The 10″ engine can be run alone by closing both *P* and *Q*.

Condensing arrangements.—A diagram of the condensing arrangements is shown in Fig. 296. *A* is a surface condenser into which the exhaust steam is led through the pipe *B*. An air pump *C* draws air and water from the condenser through the pipe *D*. The air pump is independently driven by a steam cylinder, and discharges the water through a pipe *E* into a measuring tank *F*, the temperature being measured by a thermometer at *G*. Circulating water is supplied to the condenser through a pipe *H* fitted with a regulating valve, and is discharged through the pipe *K* into a long

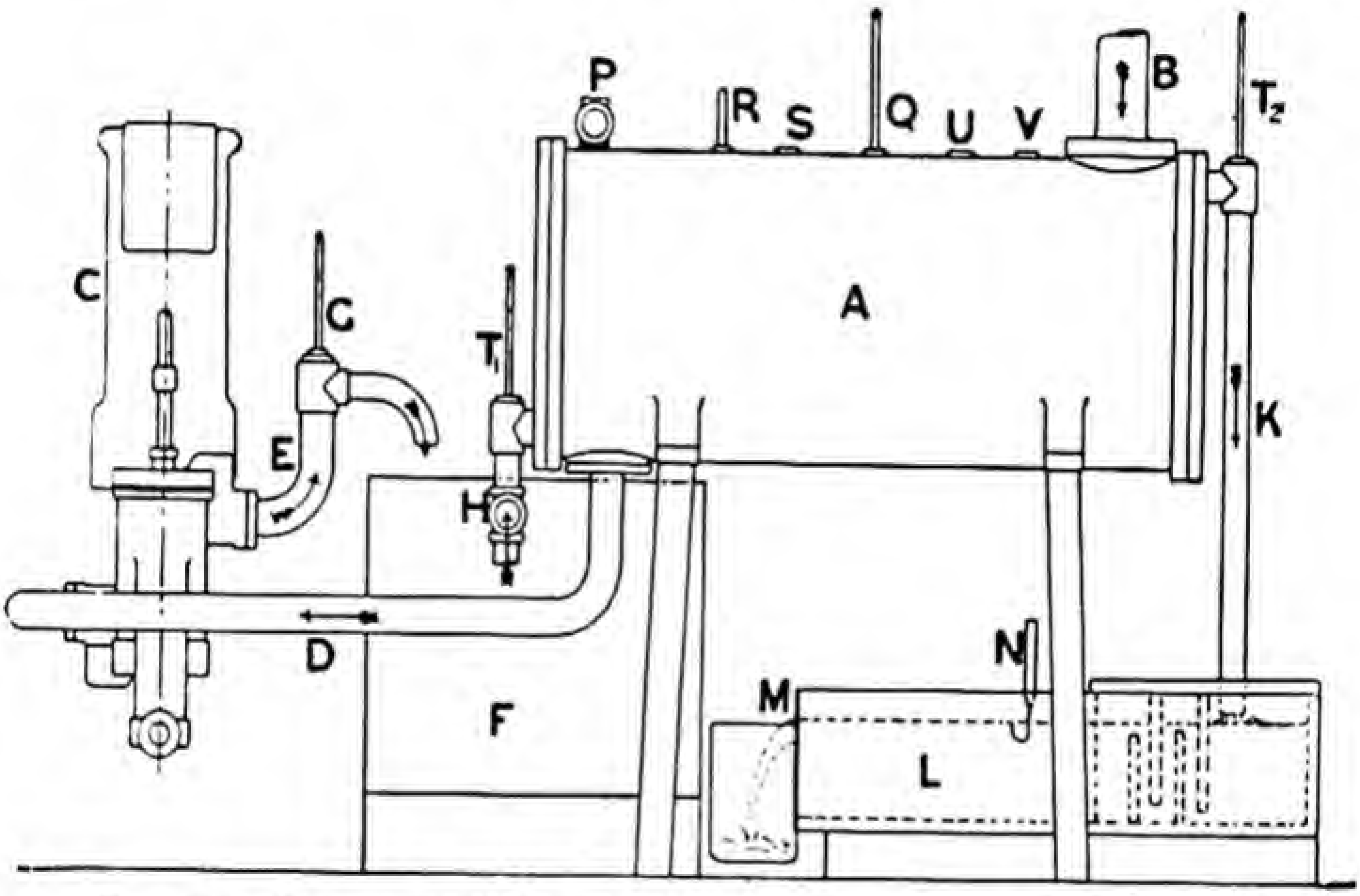


FIG. 296.—Diagram of the condensing arrangements at the West Ham Technical Institute.

box *L* fitted with a V gauge notch at *M*. The head of water over this notch is measured by a hook gauge at *N*, and from this the quantity of water used per minute may be calculated. The box *L* is fitted with baffle plates so as to steady the flow of water. Thermometers are fitted at T_1 and T_2 for measuring the inlet and outlet temperatures respectively.

A valve *P* on the top of the condenser enables the vacuum to be spoiled and the condenser worked at atmospheric pressure. The engine can thus be arranged to work against a back pressure a little in excess of that of the atmosphere, and the steam used can be measured by condensing it in the condenser. A thermometer cup at *Q* enables the temperature of the condenser to be obtained

The pipe R leads to a vacuum gauge; another pipe connects S to a mercury column used for testing vacuum gauges, the gauges being mounted at U and their readings compared with those of the mercury column. Steam may be blown into the condenser from the main steam pipe, through a pipe connected to V.

The engine driving the air pump is also fitted with a simple pipe form of surface condenser in order that the steam used in driving the air pump may be measured. This condenser works at atmospheric pressure.

The principal dimensions of the machinery are given in the following table. For convenience the 6″ and 10″ cylinders are described respectively by H.P. and L.P.

DIMENSIONS OF EXPERIMENTAL ENGINES.

	H.P.		L.P.
Diameter of cylinders,	6″		10″
Stroke,	8″		8″
Diameter of piston rods,	$1\frac{3}{8}$″		$1\frac{3}{8}$″
Travel of main valves,	2″		2″
Travel of expansion valves,	$2\frac{3}{16}$″		$2\frac{3}{16}$″
Diameter of flywheel,	3′ 0″		3′ 0″
Number of tubes in condenser,		115	
External diameter of tubes,		$\frac{3}{4}$″	
Length between tube plates,		3′ 6″	
Total cooling surface,		79 sq. feet.	
Diameter of air pump,		$5\frac{1}{4}$″	
Stroke of air pump,		5″	
Capacity of receiver,		630 cubic inches.	

B.H.P. constants:

Circumference of wheel at rope centre, - 9·55 feet.

$$\text{B.H.P.} = 0{\cdot}000289 \times \text{net pull} \times \text{revs. per min.}$$

I.H.P. constants:

	H.P.		L.P.	
	Top.	Bottom.	Top.	Bottom.
I.H.P. $= p_m \times N \times$	0·000575	0·00054	0·00158	0·00155

Tests of the engines.—The student should first have some practice in taking indicator diagrams and working them out; also

in adjusting and reading the brake loads and in measuring the speed by reading the revolution counter (p. 104). A test may then be carried out for ascertaining the mechanical efficiency of the engine under given conditions.

EXPT. 32.—The method consists in measuring the I.H.P. and B.H.P. of the engine during a run of from $\frac{3}{4}$ to 1 hour in duration. Readings should be taken every 5 minutes of the pressure of the steam supply, the revolution counter and the brake loads, and diagrams should be taken every 5 minutes. The method of recording the measurements and working out the results will be understood from the following record of a test. The student should record his own results on the same plan.

TRIAL OF STEAM ENGINE AT WEST HAM TECHNICAL INSTITUTE.

Date, - - - - - -	17th July, 1903.
Engine tested, - - - -	6″ × 8″ single cylinder.
Conditions of running, - -	non-condensing, unjacketed.
Intended steam pressure by gauge, -	50 lbs. per sq. inch.
Intended revolutions per minute, -	270.
Cut-off as a fraction of stroke, top, -	0·2.
,, ,, ,, ,, bottom,	0·28
Object of test, - -	to determine the mechanical efficiency under the given conditions.

LOG OF TRIAL 1, SERIES *C*.

17*th July*, 1903.

Time. A.M.	No. of indicator diagram.	Pressure of steam by gauge. lbs. per sq. in.	Rev. counter.	Revs. per min.	Brake loads.	
					S lbs.	*W* lbs.
10.45	1	50	9070	271	16	61
10.50	2	50	10427	269	16	61
10.55	3	50	11774	272	16	61
11.0	4	50	13133	272	16	61
11.5	5	50	14493	271	16	61
11.10	6	50	15846	270	16	61
11.15	7	50	17197	272	16	61
11.20	8	50	18559	271	16	61
11.25	9	50	19916	271	16	61
11.30	10	50	21273		16	61

Duration of trial, 45 minutes.

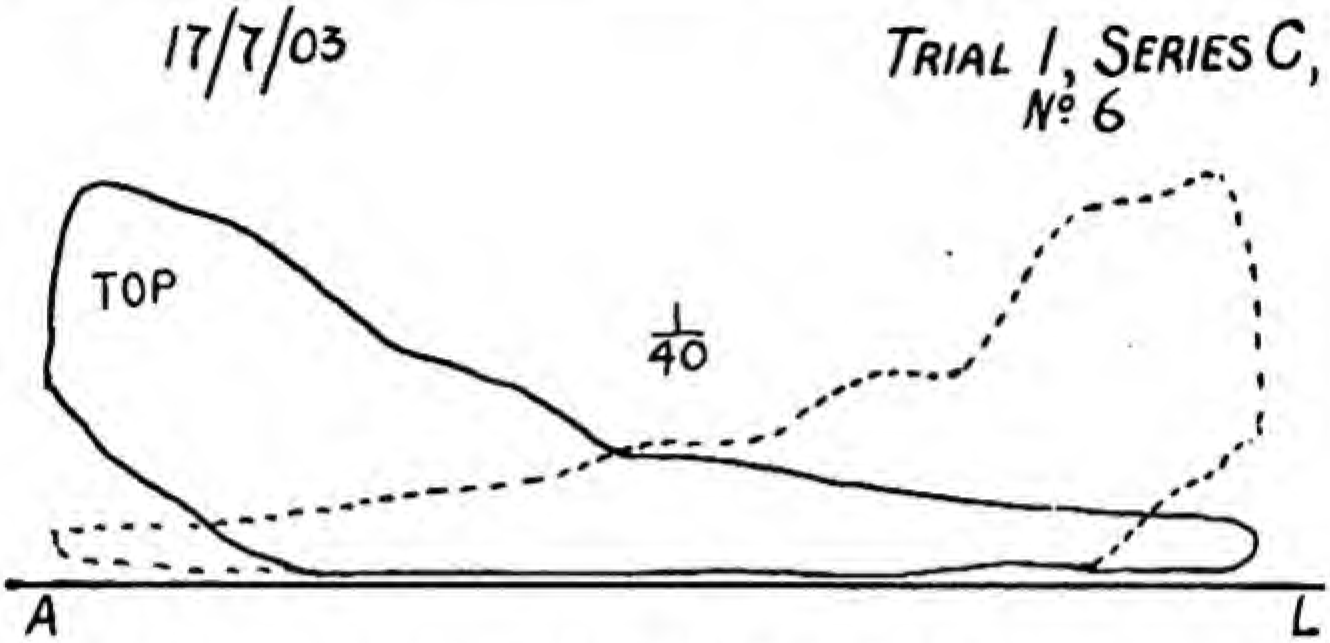


Fig. 297.—Indicator diagram from the 6″ cylinder of the experimental steam engines at the West Ham Technical Institute.

RESULTS OF THE TRIAL.

Average steam pressure by gauge = 50 lbs. per sq. inch.
Average net brake load = $W - S$ = 45 lbs.
Average revs. per min. = 271.
From the diagrams, p_m (top) = 16·8 lbs. per sq. inch.
p_m (bottom) = 16·8 lbs. per sq. inch.
I.H.P. (top) = 2·62.
I.H.P. (bottom) = 2·46.
Total I.H.P. = 5·08.
B.H.P. = 3·53.

$$\text{Mechanical efficiency} = \frac{3{\cdot}53}{5{\cdot}08} = 0{\cdot}695.$$
$$= \underline{69{\cdot}5} \text{ per cent.}$$

EXPT. 33.—Using the same method, determine the mechanical efficiency of the 10″ × 8″ engine.

EXPT. 34.—The engine being arranged to run coupled compound, with a brake on each flywheel, determine the I.H.P., B.H.P. and mechanical efficiency.

The records may be conveniently made by following the procedure of the preceding trial.

Condenser trials.—Having become familiar with the management of the indicator and brake, one section of the students may run a trial on the condenser while the others are testing the steam engines for I.H.P. and B.H.P.

EXPT. 35.—The measurements to be taken are:

(*a*) The quantity of steam used by the engine during the trial. This is obtained by noting the quantity of water discharged by the air pump into the measuring tank, readings being taken every 5 minutes. If the engine is running at very low power, the small quantity of discharged water of condensation is conveniently and accurately measured by catching it in buckets and noting the weight and the time each bucket is removed.

(*b*) The temperature of the water discharged from the air pump; obtained from 5 minute readings of the thermometer inserted in the discharge pipe.

(*c*) The temperature of the condenser, obtained every 5 minutes from the thermometer in the top of the condenser.

(*d*) The pressure in the condenser by vacuum gauge, and the barometric reading. Read the gauge every 5 minutes and the barometer at the beginning and the end of trial.

(*e*) The quantity of circulating water used per hour. Take readings of the hook gauge to give the head of water in the water channel every 5 minutes and average them. Then, for a right-angled V notch,

$$Q = 2{\cdot}635(h)^{\frac{5}{2}}.$$

Where Q = cubic feet flowing per second,

h = head of water over notch in feet.

(*f*) The inlet and outlet temperatures of the circulating water; obtained by reading the thermometers in the inlet and discharge pipes every 5 minutes.

These may be conveniently recorded thus:

TRIAL OF STEAM ENGINE AT ....

CONDENSER LOG.

Date..................

Time.	Quantity of steam condensed.	Temp. of hot well.	Temp. of condenser.	Vacuum gauge.	Barometer.	Circulating water per sec.	Circulating water temperatures. Inlet.	Circulating water temperatures. Outlet.
	lbs.	F. or C.	F. or C.	Inches of mercury.	Inches of mercury.	Cub. ft.	F. or C.	F. or C.

Results.

(1) Duration of trial, - - - - - - - ....

(2) Total I.H.P. of engine, - - - - - - ....

(3) Total B.H.P. of engine, - - - - - ....

(4) Quantity of steam condensed during trial, lbs., - ....

(5) ,, ,, ,, per hour, lbs., - ....

(6) Steam used per I.H.P. per hour, lbs. $=\frac{(5)}{(2)}$, - - ....

(7) ,, ,, ,, B.H.P. ,, lbs. $=\frac{(5)}{(3)}$, - - ....

(8) Average temperature of hot well (F. or C.), - ....

(9) Average temperature of condenser (F. or C.), - ....

(10) Average absolute pressure in condenser, lbs. per sq. inch, - - - - - . - ....

(11) Circulating water per hour, lbs., - - - - ....

(12) Average inlet temp. of circulating water (F. or C.), ...

(13) ,, outlet ,, ,, ,, (F. or C.), ...

(14) Circulating water per hour per I.H.P. $=\frac{(11)}{(2)}$, - ....

(15) Cooling surface of condenser, square feet, - - ....

(16) Steam condensed per hour per sq. foot of cooling surface, lbs. $=\frac{(5)}{(15)}$, - - - - - - ....

(17) Circulating water per hour per sq. foot of cooling surface, lbs. $=\frac{(11)}{(15)}$, - - - - - - ....

Additional calculations on the condenser.—The following calculations are of great importance and should be worked carefully by the student.

(*a*) Calculate the **heat given up in the condenser by the exhaust steam,** stating the result in B.T.U. per hour. First find from the Tables, p. 454, the total heat of a pound of steam at the absolute pressure and temperature of the condenser given by (10) and (9) above. The sensible heat carried away in the condensed water passing through the air pump may be calculated from result (8) above. Thus,

Let H = total heat in B.T.U. of exhaust steam,

t_h = temp. F. of hot well.

Then, sensible heat carried away per lb. water $= t_h - 32$;

heat given up in condenser per lb. water $= H - (t_h - 32)$

$= H - t_h + 32$, B.T.U.

It will be noted that it is assumed that the exhaust steam entering the condenser is dry saturated.

From result (5) above the total heat given up per hour may be calculated.

Let W_s = steam condensed per hour, lbs.

Then,

Heat given up in condenser per hour $= (H - t_h + 32)\,W_s$, B.T.U.

(*b*) Calculate the **heat taken up by the circulating water passing through the condenser,** stating the result in B.T.U. per hour. Using results 11, 12 and 13 above,

Let, W_c = circulating water per hour, lbs.,

t_1 = inlet temperature, F.,

t_2 = outlet temperature, F.

Then,

Heat taken up per hour by circulating water $= W_c(t_2 - t_1)$, B.T.U.

Supposing no heat to be wasted in the condenser, the result of calculation (*a*) should be equal to that of calculation (*b*). If not, try to explain the discrepancy.

(*c*) During the trial the engine was supplied with W_s lbs. weight of steam per hour at a pressure p_1 shown by the gauge attached to the steam supply pipe. The total heat in one pound weight of this steam may be obtained from the Table, p. 454. Let H be this quantity in B.T.U.

Heat has been carried away through the air pump by each pound weight of water to the extent of

$$(t_h - 32),\ \text{B.T.U.}$$

Therefore each pound weight of steam passing through the engine has parted with heat given by

$$H - (t_h - 32),\ \text{B.T.U.};$$

Total heat parted with per hour by the steam

$$= W_s\{H - (t_h - 32)\},\ \text{B.T.U.} \quad \ldots\ldots\ (1)$$

This heat represents the energy available for the production of mechanical work. To obtain the actual amount of the latter for the calculated expenditure of heat:

Work done on the piston per min. = I.H.P. × 33,000,

" " " " " hour = I.H.P. × 33,000 × 60 ft.-lbs.

$$= \frac{\text{I.H.P.} \times 33{,}000 \times 60}{J},\ \text{B.T.U.} \quad \ldots\ldots\ (2)$$

To obtain the **efficiency of the machine as a heat engine**, divide the work done on the piston per hour by the heat parted with per hour by the steam, both in B.T.U., giving:

$$\text{Efficiency} = \frac{\text{I.H.P.} \times 33{,}000 \times 60}{W_s\{H-(t_h-32)\}J}.$$

Willans's law.—The late Mr. Willans made many series of experiments on his high-speed engines with very valuable results.

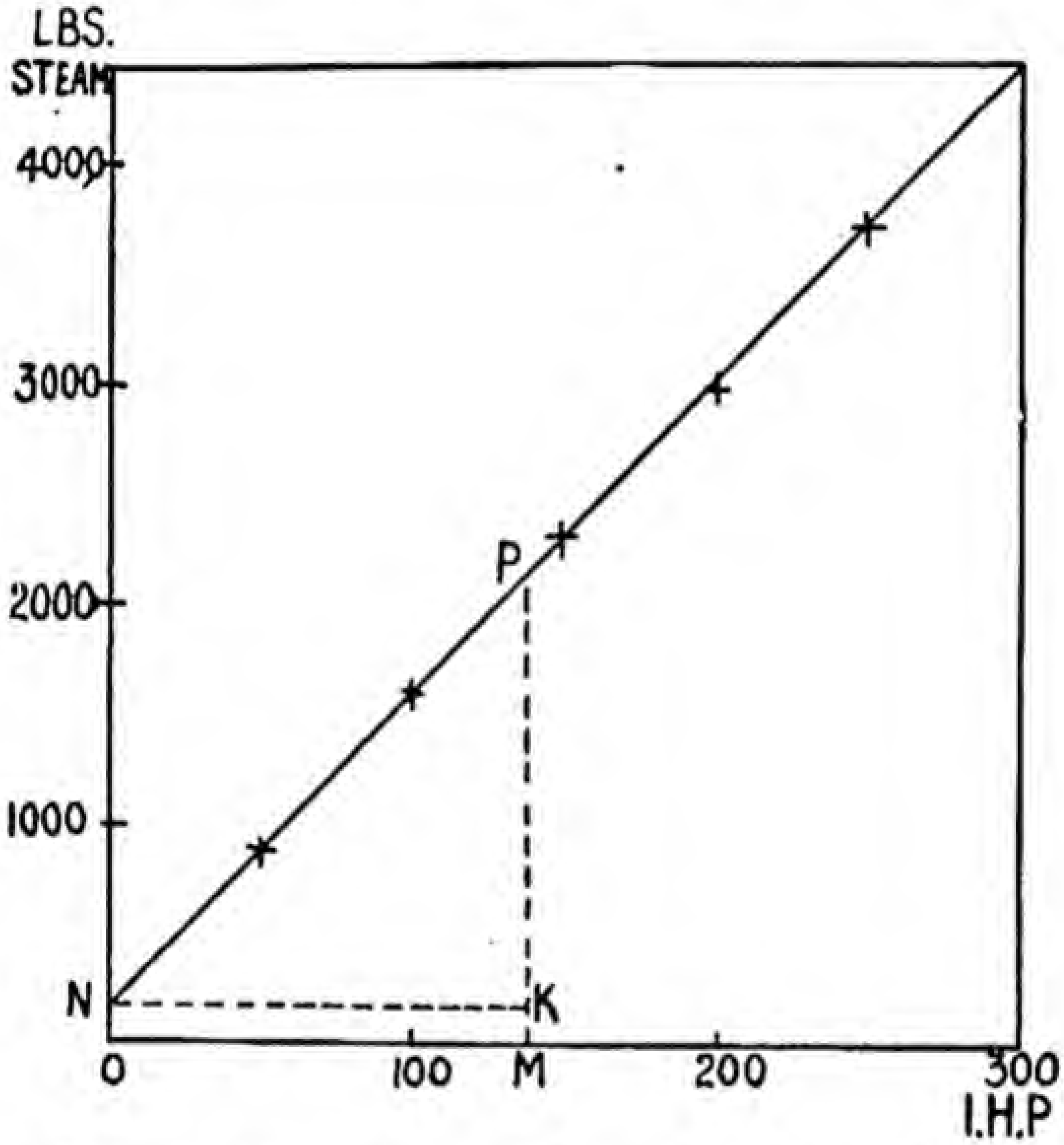


FIG. 298.—Diagram illustrating Willans's straight-line law.

Among other principles deduced from these results is the law that **in a given engine with constant cut-off, but H.P. varied by varying the steam pressure, the curve produced by plotting total steam consumption per hour against I.H.P. or B.H.P. will be a straight line.** The curve shown in Fig. 298 has been plotted from data obtained from tests under these conditions, giving the straight line *NP*.

The law may be expressed algebraically. Notice that the steam consumption, Q, for a given I.H.P. represented by OM is given by PM, and that PM is made up of two quantities, MK and PK. PK

is in a constant ratio, a, to the I.H.P., while MK is constant for all powers and is equal to ON. We may therefore write:

$$PK = a \times \text{I.H.P.},$$

$$MK = a \times b,$$

$$Q = PM = PK + MK,$$

$$Q = a \times \text{I.H.P} + a \times b$$

$$= a(\text{I.H.P.} + b).$$

In this a will be the steam consumption per I.H.P. over and above a certain constant quantity which the engine cylinder consumes at all powers without the production of work, represented by the product of the constants a and b in the above results.

The law for B.H.P. and Q, the total steam consumption per hour, may be written in a similar manner, giving

$$Q = c(\text{B.H.P.} + d),$$

in which c is the constant steam consumption per hour per B.H.P. over and above a certain constant quantity cd, which is expended at all powers in driving the engine itself.

The Willans's curve obtained by plotting horse-power and total steam consumption per hour under the conditions of uniform pressure of steam supply and varied ratio of expansion is not a straight line but a curve which tends to become steeper at higher powers.

Tests on steam boilers.—In simple tests of steam boilers for efficiency, the following measurements are taken.

(a) **Weight of coal supplied per hour.** This is obtained by supplying a measured quantity, usually 50 lbs., to the stoker, recording time and quantity, and supplying a further equal quantity when the first has been used, again noting the time. This procedure carried out throughout will enable the rate of combustion to be ascertained at any part of the trial.

(b) **The total feed water supplied and evaporated during the trial.** In carrying out a boiler trial it is very necessary to preserve the conditions of working as uniform as possible. To enable this to be done, the rate of drawing steam away from the boiler should be kept uniform by running the engines or other plant at a steady load. The feed should also be supplied uniformly, which may be accomplished by regulating the supply so as to

preserve a constant water level in the boiler. To facilitate this, a piece of fine cord should be tied round the water-gauge glass at the water level when starting the trial. The feed water is best measured by means of tanks. In Fig. 299, *A* is the main feed supply tank, furnished with a cock *B* for supplying water to a measuring tank *C*. A scale graduated in lbs. of water is placed in *C*. When measured, the water is discharged through a cock *D* into a supply tank *E* from which the feed pump draws its supply of water through the pipe *F*. The tank *F* has a pointer *G* fixed in it so that the water may be brought to a standard level.

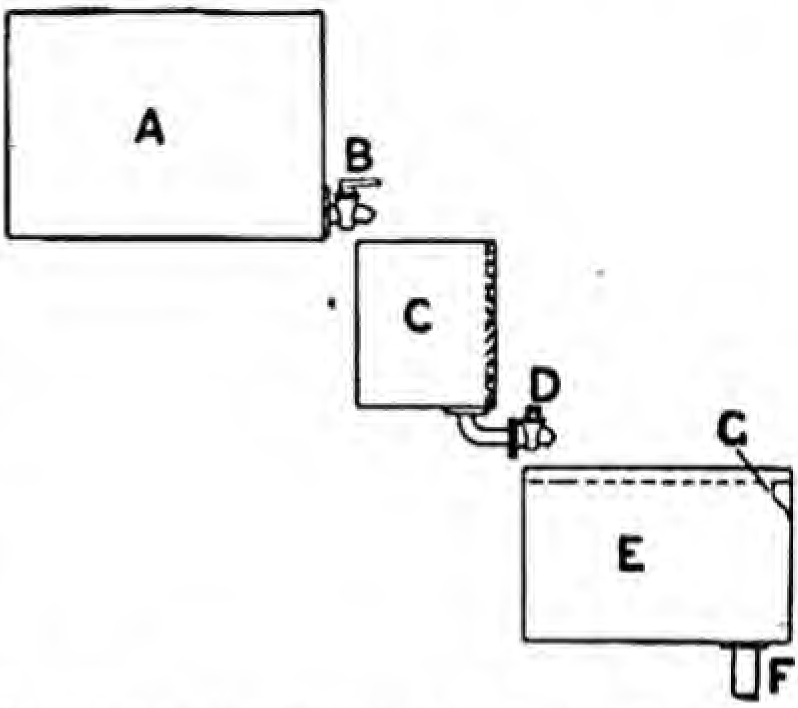


FIG. 299.—Arrangement of tanks for measuring the feed water.

At the beginning of the trial, the water in the boiler is at the level of the cord on the gauge glass, and water is run into *E* up to the standard level. During the trial, water is supplied to *E* from the measuring tank in uniform quantities by filling *C* up to say 80 lbs., then closing cock *B* and opening *D*, the time of opening the latter and also the quantity being recorded. At the end of the trial, the water in the boiler and also in *E* are brought to the same standard levels, consequently all water which has been measured in the tank *C* has been passed by the feed pump into the boiler. Note that if an injector is used for feeding the boiler, any overflow from the injector should be weighed and deducted from the total water measured.

The following should be recorded every 10 minutes:

(*c*) Steam pressure by gauge. This should be kept steady, the stoker being made responsible.

(*d*) Temperature of feed water supplied; measured by a thermometer in the tank *E* (Fig. 299).

(*e*) Temperature inside the boiler house.

(*f*) Temperature outside the boiler house.

Starting and finishing a trial.—To obtain definite information regarding the performance of a given boiler, the test must be extended over from 6 to 10 hours. This is rendered necessary by the uncertainty of the furnace conditions at the start and finish of the trial. In some important trials, the plan is adopted of shutting down the stop valve of the boiler just before starting and cleaning fires and ashpits. A measured quantity of coal is then allowed the stoker in order to start his fire, on completion of which the test is started. Otherwise, the state of the fire at the beginning and end of the test may be roughly guessed to be alike. In either case, the ashpit ought to be cleaned before starting, in order that its contents may be weighed at the end of the trial in order to determine the approximate quantity of ash in the coal burned. The uncertainty of the state of the furnaces at the start and finish introduces an error in the estimated coal consumption which will be of small percentage provided the trial has been of sufficient duration.

Graphical log.—The conditions throughout the trial, as has been stated, should be preserved as nearly uniform as possible. A useful way of checking these consists in plotting the whole of the observations and time on the same sheet, using different colours for the various sets of measurements. A glance at the resulting diagram will give the desired information regarding the uniformity of the conditions. Thus, the coal supply and also the water supply will be represented by sloping lines which should be nearly straight, the plotting of these being effected by taking the total quantities supplied up to the time which is being plotted. The boiler pressure line should be nearly straight and horizontal. The variation of other conditions, such as temperatures, over which there is little or no control, will be shown in the diagram.

Example of a trial.—To illustrate the method of recording and working out the results, the following example is included. Students should adopt a similar method in recording their own results.

TRIAL OF A BABCOCK & WILCOX WATER-TUBE BOILER AT WEST HAM TECHNICAL INSTITUTE.

Date, 15TH JUNE, 1906.

The object of the trial was to determine the quantity of water evaporated per pound of coal while supplying steam to the experimental engines.

Log of Trial.

Time.	Pressures.			Temperatures.			Remarks.
	Boiler gauge.	Chimney draft.	Barometer.	Outside house	Inside house.	Feed supply, tank.	
	lbs. per sq. in.	inches water.	cms. mercury.	F.	C.	C.	
11.30	84	$\frac{3}{8}$	76·3	55	19	17·8	Stoker A.
11.40	80	,,		54½	19	17·8	Started
11.50	82	,,		55	18½	17·8	with clean
Noon							ashpits.
12.0	78	,,		54½	18	18·0	
12.10	80	,,		54	18	18·0	
12.20	79	,,		55	18	18·0	
12.30	80	,,		56	18	18·0	
12.40	80	,,		55	18	18·0	
12.50	80	,,		55	18½	18·0	
1.0	80	,,		57	18	18·0	Stoker B.
1.10	80	,,		56	19	18·0	
1.20	81	,,		56	19	18·0	
1.30	80	,,	76·27	57	19	18·0	
1.40	81	,,		58	19	18·0	
1.50	77	,,		58	19	18·0	
2.0	80	,,		59	19	17·6*	*Ball valve
2.10	82½	,,		58	19	17·7	opened,
2.20	80	,,		58	19	17·0	admitting
2.30	80	,,		59	19	17·0	fresh cold
2.40	80	,,		59	20	16·8	supply to
2.50	80	,,		60	20	17·0	tank.
3.0	82	,,		60	20	17·0	Stoker A
3.10	81	,,		59	21	17·0	to end.
3.20	80	,,		57	20	17·0	
3.30	80	,,	76·26	56½	19½	17·1	
3.40	80	,,		57½	20	17·2	
3.50	82	,,		57½	20	17·3	
4.0	80	,,		56	20	17·2	
4.10	80	,,		56	19½	17·4	
4.20	80	,,		57	19½	17·4	
4.30	80	,,	76·22	57	20	17·4	Contents of
4.40	79	,,		56	20	17·4	ashpit—
4.50	76	,,		56½	20	17·4	15½ lbs. at
5.0	80	,,		56	19½	17·5	end.
Averages	80	$\frac{3}{8}$	76·26	57° F.	19° C. = 66° F.	17½°C. = 63½° F.	

Fuel Supply.		Feed Water.		Remarks.
Time.	Lbs. supplied.	Time.	Lbs. supplied.	
11.41 A.M.	100	11.33	320	
12.44 P.M.	100	11.58	320	
1.38	50	12.14	320	
2.4	50	12.29	320	
2.40	50	1.5	320	
3.20	50	1.36	320	
3.46	50	1.50	320	
4.25	50	2.14	320	
		2.30	320	
		2.53	320	
		3.14	320	
		3.39	320	9 lbs. coal returned
		4.10	320	unused at end.
		4.37	320	
		5.0	480	
Totals.	491		4960	

The water was fed to the boiler by means of an injector. The temperature of the water in the injector delivery pipe varied from 133° to 135° F. ; the average temperature of the feed delivered is taken as 134° F.

Results.

Duration of trial, - - - - - - $5\frac{1}{2}$ hours.
Total feed water evaporated, - - - 4960 lbs.
Total coal burned, - - - - - - 491 lbs.
Feed water evaporated per lb. of coal, - $\frac{4960}{491} = 10{\cdot}1$ lbs.
Total heat of steam at 80 lbs. per sq. inch gauge, 1177 B.T.U.
Sensible heat in 1 lb. of feed water as delivered, $(134 - 32) = 102$ B.T.U.
Heat given in boiler to each pound of feed water, - - - - - - - $(1177 - 102) = 1075$ B.T.U.

Equivalent evaporation $= 10{\cdot}1 \times \frac{1075}{967} = 11{\cdot}22$ lbs. water from and at 212° F. per lb. of coal.

Percentage ash $= \frac{15\frac{1}{2}}{491} \times 100 = 3{\cdot}2$.

Dryness of steam.—Steam from a boiler working without a superheater always contains some moisture. Usually it is found that, in every 100 lbs. of mixture which have just passed from the boiler into the steam pipe, there are from 2 to 5 lbs. of water, both steam and moisture being at the same temperature. This fact is important in its bearing on the heat imparted to the water in the boiler. Thus, the steam in the mixture has taken up both latent and sensible heat, while the water has taken up sensible heat only.

Supposing that in 1 lb. of the mixture there are q lb. of steam and $(1-q)$ lb. of water; q is called the **dryness** fraction.

Let L = the latent heat and h = the sensible heat of dry saturated steam at boiler pressure. Then in each pound of the mixture there will be heat given by:

$$\text{Heat} = qL + h.$$

This latter amount is the quantity of heat which should be used in the calculation for the equivalent evaporation instead of the total heat of dry saturated steam.

Testing for the dryness of steam.—There are two methods for determining the dryness fraction of steam. In the first a **separating calorimeter** is used; in the second, a **throttling calorimeter.** The M'Innes instrument combines these methods, and either instrument may be used alone or both in series.

Steam is drawn from the steam pipe A, Fig. 300, through a sampling tube perforated with slots, and after passing through a regulating valve B enters the separating calorimeter D. Here the greater part of the water clings to the walls, trickles to the bottom, and, when enough has collected as shown by the gauge glass C, it is drawn off through the valve E to be weighed. The nearly dry steam escapes through a side pipe near the top of D into the throttling calorimeter. Here it is discharged through a small orifice in a plate H, which is heat-insulated by vulcanite joints G, G. Thermometer pockets are provided at F and F' in order that the temperature of the steam, before and after passing the orifice, may be measured. The steam is led finally through a coil (not shown) immersed in a vessel of cold water so that it may be condensed; the resulting water is carefully weighed.

The action in the throttling calorimeter depends on the facts that practically no work is done by the steam in passing the orifice, although its pressure falls to nearly atmospheric, and also that the

total heat of dry saturated steam at the higher pressure before the orifice is greater than that of dry saturated steam at the lower pressure of discharge. As practically no work has been done, it follows that the steam after passing the orifice will, if not at first too wet, have heat in excess of that required for dry saturated steam at the lower pressure, and consequently will become super-

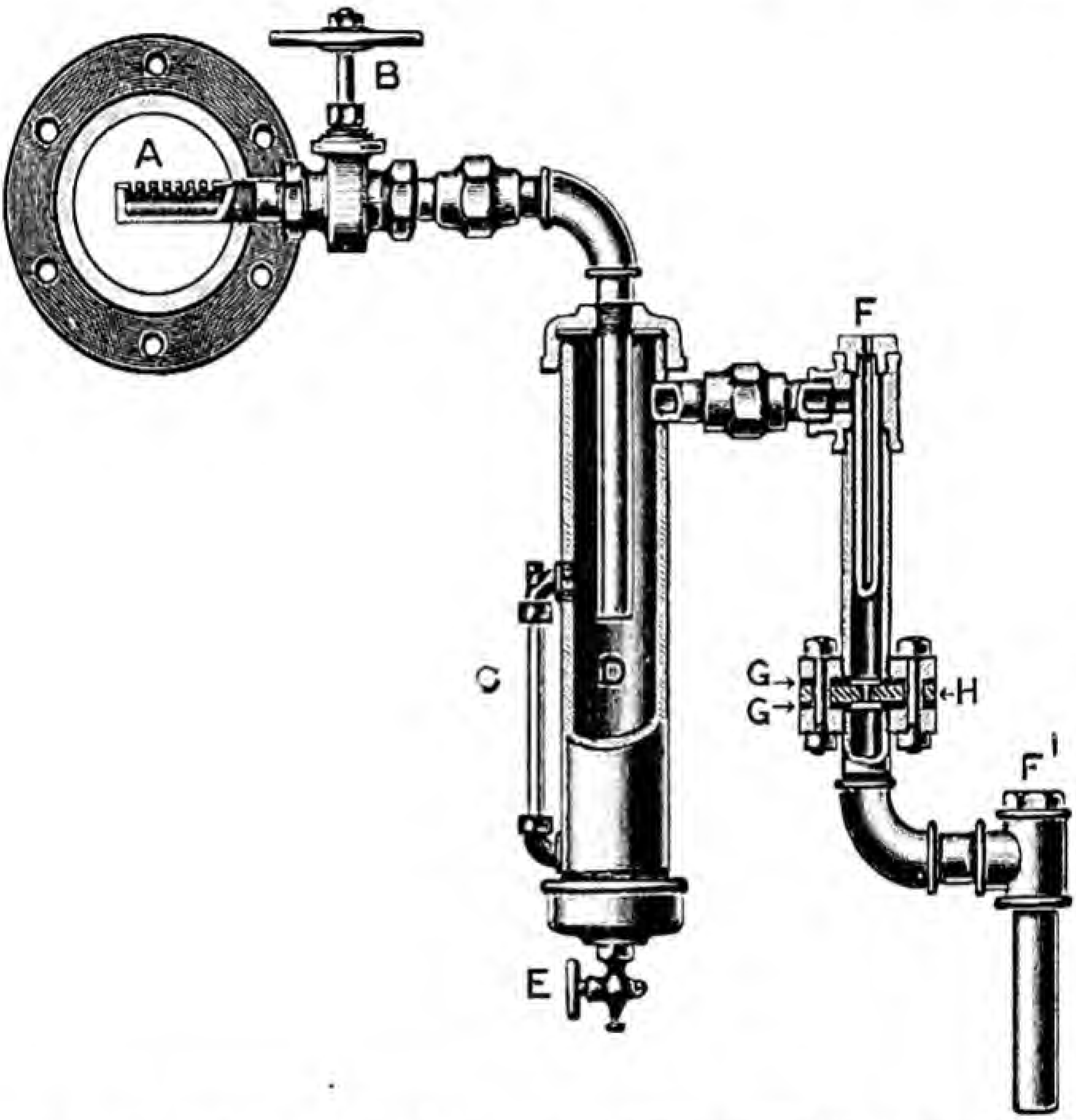


FIG. 300.—M'Innes combined separating and throttling steam calorimeter.

heated. These principles enable the dryness fraction to be calculated from the measurements taken.

The throttling calorimeter may be disconnected, and the separator connected direct to the cooling coil and used alone; or the throttling calorimeter may be connected direct to the valve *B* and used alone.

EXPT. 36.—To carry out a test with the separating calorimeter, which is used for very wet steam only, allow the steam to blow

through for a few minutes in order to warm the instrument. Then let the test go on until sufficient water has collected to be drawn off through E, at the same time allowing the water from the condensing coil to discharge into a vessel.

Let W_w = weight of water, in lbs., drawn off through E,
W_s = weight of water, in lbs., passed through the coil.

Then

$W_w + W_s$ = total weight of sample drawn from A,
and W_s = weight of steam in the sample;

$$\therefore \text{ dryness fraction} = \frac{W_s}{W_w + W_s}.$$

EXPT. 38.—To carry out a test with the throttling calorimeter alone, the condensing coil is not required, the steam being allowed to discharge freely. Let the steam discharge freely through the instrument for 10 minutes so as to warm it thoroughly and to allow the conditions to settle down. Then read the temperatures shown by the thermometers in F and F'.

Let t = temperature Fah. at F,
t' = temperature Fah. at F'.

From the Table, p. 454, find

L = latent heat of steam at temperature t° Fah.;
h = sensible heat of steam at temperature t° Fah.

Let q = the dryness fraction.

Then

Heat per lb. of stuff before reaching the orifice $= qL + h$. .... (1)

The pressure after passing the orifice may be taken as 16 lbs. per square inch absolute, corresponding to a temperature of 216·3° Fah. The total heat of saturated steam at this temperature will be

$$H_2 = 1147 \cdot 9 \text{ B.T.U.}$$

If the steam has been superheated, it will be found that t' is higher than 216·3° Fah., the degree of superheat will be $(t' - 216 \cdot 3)$.

Assuming that the specific heat of steam is 0·48 and is constant,[1]

Heat used in superheating 1 lb. of the steam $= 0 \cdot 48(t' - 216 \cdot 3)$ B.T.U.,

$\therefore$ Total heat in 1 lb. of steam after passing the orifice

$$= 1147 \cdot 9 + 0 \cdot 48(t' - 216 \cdot 3) \text{ B.T.U.} \qquad (2)$$

[1] The more correct number is 0·6 (p. 71); the older value is assumed in this calculation as curves published for use with throttling calorimeters are generally based on this value.

As practically no heat is lost, it follows that (1) and (2) will be equal, giving

$$qL + h = 1147{\cdot}9 + 0{\cdot}48(t' - 216{\cdot}3)$$

$$q = \frac{1147{\cdot}9 + 0{\cdot}48(t' - 216{\cdot}3) - h}{L} \quad \text{.................. (3)}$$

from which the dryness fraction may be calculated.

If the instruments are used in series, as shown in Fig. 300, the condensing coil must be used, and the experiments carried out simultaneously on each instrument as above directed. The total moisture will be the sum of the results obtained from each instrument.

EXAMPLE. In an experiment with the instruments in series, the following quantities were obtained:

Separating calorimeter, $W_s = 2{\cdot}1$ lbs.

$W_w = 0{\cdot}37$ lb.

$$q_s = \frac{2{\cdot}1}{2{\cdot}1 + 0{\cdot}37} = 0{\cdot}85$$

$= \underline{85}$ per cent.

Moisture present 15 per cent.

Throttling calorimeter, $t = 338°$ F.

$t' = 250°$ F.

$L = 876{\cdot}3$ B.T.U.

$h = 308{\cdot}7$ B.T.U.

$$q_t = \frac{1147{\cdot}9 + 0{\cdot}48(250 - 216{\cdot}3) - 308{\cdot}7}{876{\cdot}3}$$

$= 0{\cdot}976$

$= \underline{97{\cdot}6}$ per cent.

Moisture present 2·4 per cent.

This represents the percentage moisture on the quantity reaching the throttling calorimeter, viz. 2·1 lbs.

$$\text{Moisture detected in throttler} = \frac{2{\cdot}1 \times 2{\cdot}4}{100} = 0{\cdot}0504 \text{ lb.}$$

$$\text{Total moisture present} = 0{\cdot}37 + 0{\cdot}0504$$

$$= 0{\cdot}4204 \text{ lb.}$$

$$\text{Percentage moisture} = \frac{0{\cdot}4204}{2{\cdot}47} = \underline{17{\cdot}02}.$$

Dryness fraction $= \underline{82{\cdot}98}$ per cent.

EXERCISES ON CHAPTER XVIII.

1. A steam engine has to be tested for mechanical efficiency. State what measurements must be taken.

2. Explain how the steam consumption of an engine running under given conditions may be measured. Supposing the boiler to be supplying steam to this engine only and that the feed water is measured. How would the quantity of feed water probably compare with the quantity of steam actually passing through the engine?

3. Circulating water at a temperature of 15° C. is supplied to a condenser and leaves with a temperature of 35° C. 950 lbs. weight of dry steam at a pressure of 3 lbs. per square inch absolute enter the condenser per hour. Hot well temperature 50° C. Find the probable quantity of circulating water per hour. Take any quantities you require from the Table, p. 455.

4. A steam engine uses 20 lbs. weight of steam per I.H.P. per hour. The steam enters the steam chest 5 per cent. wet, and at a pressure of 90 lbs. per square inch by gauge. Calculate what quantity of heat in B.T.U. is supplied per I.H.P. per hour.

5. Explain Willans's law connecting I.H.P. and steam consumption.

6. What quantities must be measured in testing a steam boiler for evaporative capacity? What precautions must be taken?

7. Describe any experiment for testing the dryness of steam. Give sketches of apparatus used.

8. Sketch and describe the construction of the air-pump of a condensing engine. What is the use of the air-pump? If the temperature of the injection water supplied to a jet condenser be 62° F., and the water is pumped out of the hot well at a temperature of 106° F., and the steam to be condensed enters the condenser at a temperature of 212° F., what weight of injection water would be required per pound of steam condensed? 1896.

9. At an Electric Light Station: on full power (or load factor 100 per cent.) the output is 6000 kilowatts, the feed water being 132,000 lbs. per hour. When the output is 1200 (or load factor 20 per cent.), the feed water is 53,000 lbs. per hour. Plot power and water on squared paper and assume a straight line law. What is the water per hour when the load factor is 10 per cent? (that is, the output is 600 kilowatts). Tabulate the numbers. State in each case the water per hour per kilowatt. 1905.

10. The total heat, that is, the heat H required to convert a pound of water at 0° C. into a pound of wet steam at θ° C., having a dryness fraction x is

$$H=\theta+xL,$$

where L is the latent heat of 1 lb. of dry saturated steam. If wet steam 90 per cent. dry (that is $x=0{\cdot}9$) at 203·3 lbs. per square inch is

throttled by passing through a non-conducting reducing valve to 101·9 lbs. per square inch, what is its dryness at the lower pressure? Remember that H is the same for the two kinds of steam; it keeps constant when steam is throttled.

p	θ	L
203·3	195	468·0
101·9	165	489·0

1906.

CHAPTER XIX.

GAS ENGINES.

Internal combustion engines.—Engines in which the combustion of the fuel is carried out in the cylinder or in vessels in direct communication with the cylinder are called internal combustion engines. The fuel employed may be solid, liquid or gaseous. The use of solid fuel, such as coal in the form of dust, is in the experimental stage at present, and there are great difficulties to be overcome before a practical engine can be put on the market. Engines using various kinds of gas and oil are very largely used and are perfectly trustworthy. Almost all of these are operated under the Beau-de-Rochas cycle, laid down in 1862 and first successfully used in the Otto silent gas engine in 1876.

Beau-de-Rochas cycle.—This cycle is carried out on one side of the piston during four consecutive strokes as follows:

1. **Charging stroke.**—During the first out-stroke of the piston, an explosive mixture of gaseous fuel and air, or of oil which has been first gasified and air, is drawn into the cylinder.

2. **Compression stroke.**—This is the in-stroke of the piston immediately following the charging stroke. During this stroke, the cylinder valves are all closed, and the explosive mixture, or charge, is compressed by the returning piston into the clearance space at the rear of the cylinder.

3. **Explosion and expansion stroke.**—At the beginning of this stroke the charge is ignited. A very rapid increase of pressure occurs, due to the heat energy of the charge being liberated by the combustion, and the piston moves forward and completes its out-stroke under the influence of the pressure of the expanding products of combustion.

4. **Exhaust stroke.**—An exhaust valve is opened a little before the end of the expansion stroke is reached, and the products of combustion at once begin to rush out of the cylinder. During the succeeding in-stroke of the piston the waste gases are driven out of the cylinder, and, at the end of this stroke, the engine is ready to take in a fresh charge.

Diagram showing the cycle.—Fig. 301, which has been drawn from an indicator diagram taken from a gas engine, will explain the Beau-de-Rochas cycle

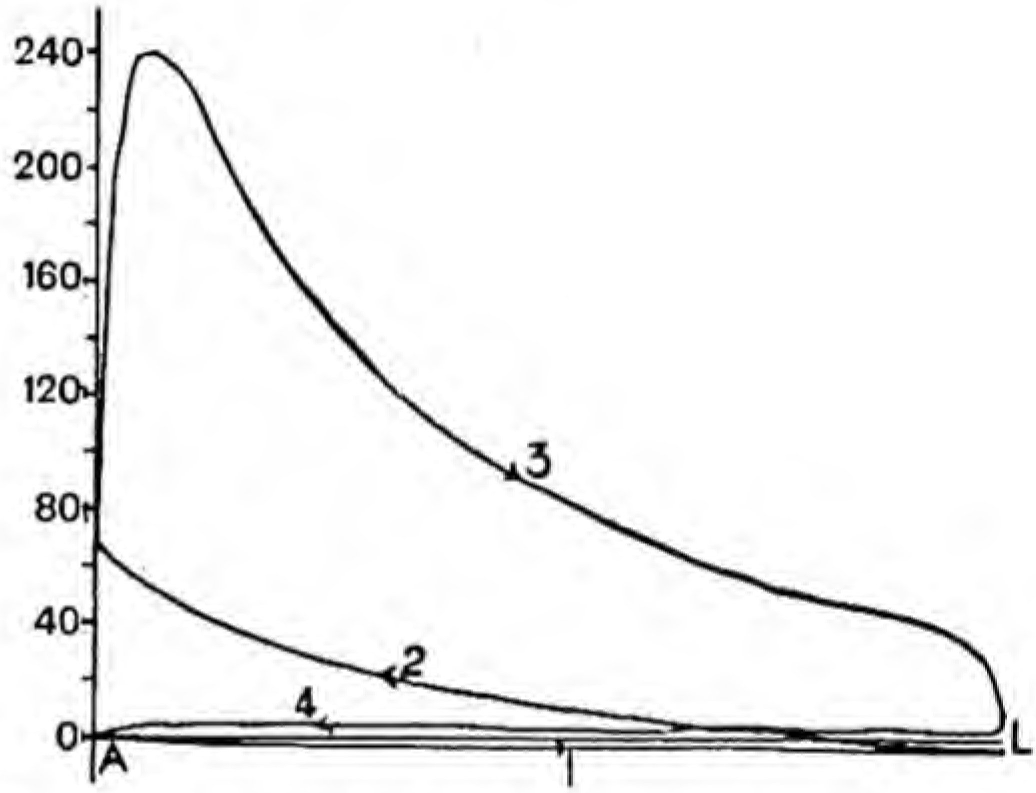


FIG. 301.—Indicator diagram showing the cycle in a gas engine.

1 is the charging stroke, during which the pressure falls slightly below atmospheric.

2 is the compression stroke, at the end of which the pressure will range from 50 to 200 lbs. per square inch above atmospheric pressure, depending on the type of engine.

3 is the explosion and expansion stroke.

4 is the exhaust stroke.

During the exhaust stroke the pressure usually rises slightly above that of the atmosphere. The charging and exhaust stroke pressures may be shown conveniently by an indicator diagram in which a light spring has been used (Fig. 302). The indicator piston rod must have a stop fitted in order to prevent the spring being broken when explosion occurs. Compression pressures of 170 to 200 lbs. per square inch, giving explosion pressures up to

350 lbs. per square inch are now being used in many large gas engines.

General description of the engine.—The cylinder in which the Beau-de-Rochas cycle is carried out is generally open at one end and only one side of the piston is used, *i.e.* the engine is single acting. The piston is connected direct to the crank by a connecting rod. The various valves for admitting the charge, exhaust valve, etc., are of the mushroom type, held down on their seats by springs, and operated by levers worked by cams on a side shaft driven from the crank shaft. As the heat developed during the combustion is large and produces a very high temperature, means must be taken for keeping the cylinder cool. Usually the cooling is effected by causing water to circulate in a water-jacket formed round the cylinder. Ignition of the charge may be effected by

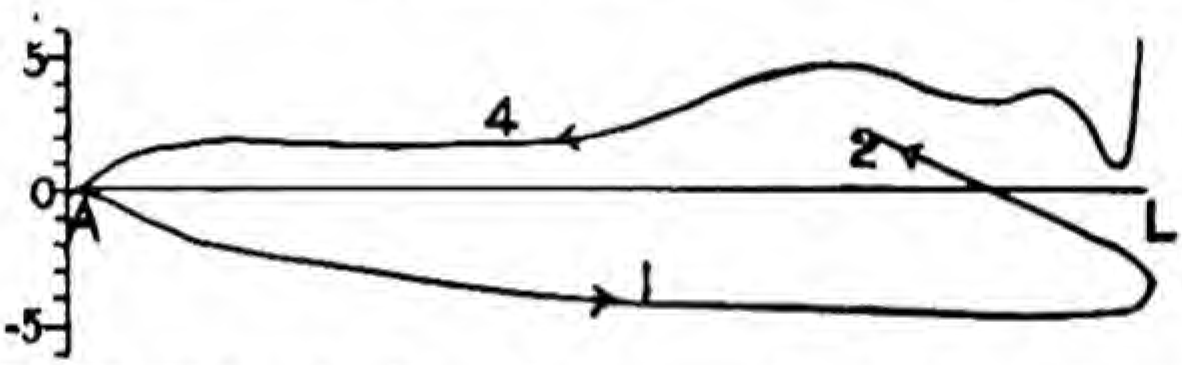


FIG. 302.—Light spring diagram from a gas engine, showing the exhaust and charging strokes.

means of a hot tube, or by an electric spark. Governing of the speed of the engine is secured by cutting off or regulating the supply of fuel to the cylinder, the governor by which this is done being generally of the centrifugal type.

Crossley gas engine.—The engine described here was constructed by Messrs. Crossley Bros. in 1898 for the Engineering Laboratory at the West Ham Technical Institute. The engine is of $6\frac{1}{2}$ B.H.P., and is intended to have a working speed of 200 revolutions per minute. The general arrangement will be understood by reference to Figs. 303, 304 and 305.

A is the cylinder, consisting of an outer casing and an inner liner, the space between serving as a water jacket, *W*. Water is supplied to the jacket through an inlet pipe *S* entering at the under side of the cylinder, and is discharged from the top through the pipe *T*. The cylinder is open at the end facing the crank shaft and is fitted with a piston *B*, made long so as to serve as a guide,

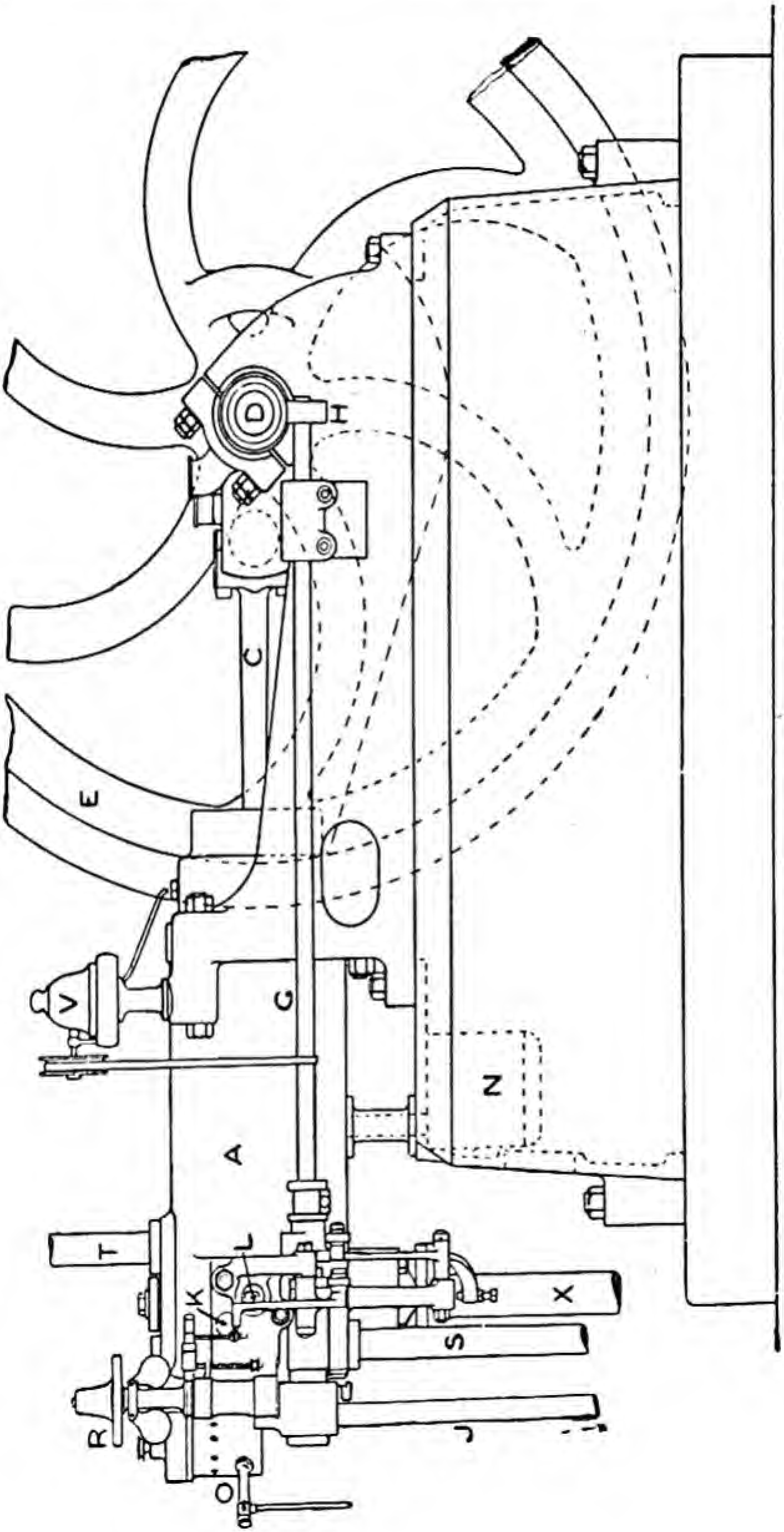


FIG. 303.—Side elevation of a 6½ B.H.P. Crossley gas engine.

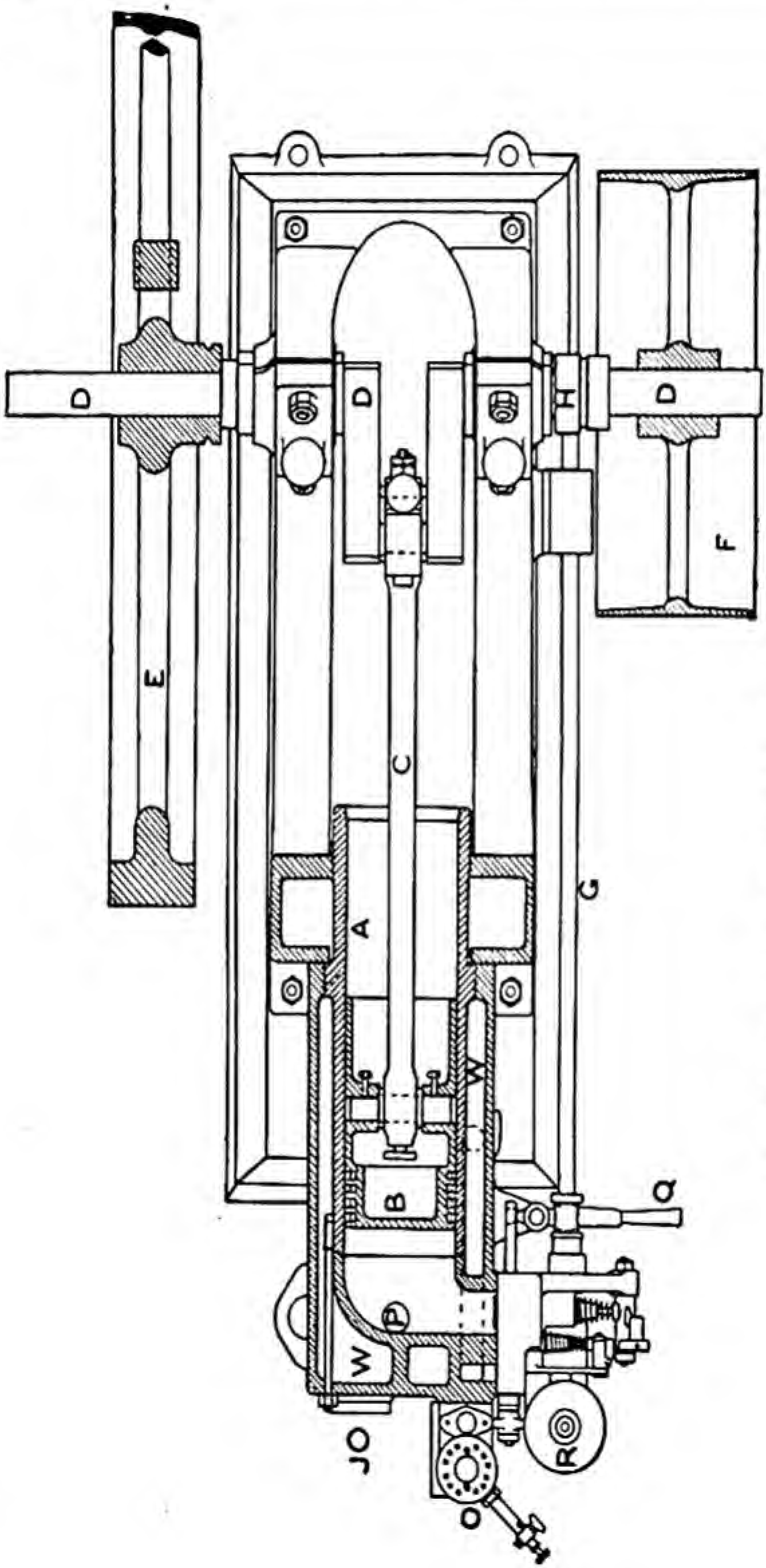


FIG. 304.—Sectional plan of 6½ B.H.P. Crossley gas engine.

and kept gas-tight by means of four spring rings fitted to grooves cut near the inner end of the piston. The connecting rod *C* is attached to the piston by means of a pin held in position by two

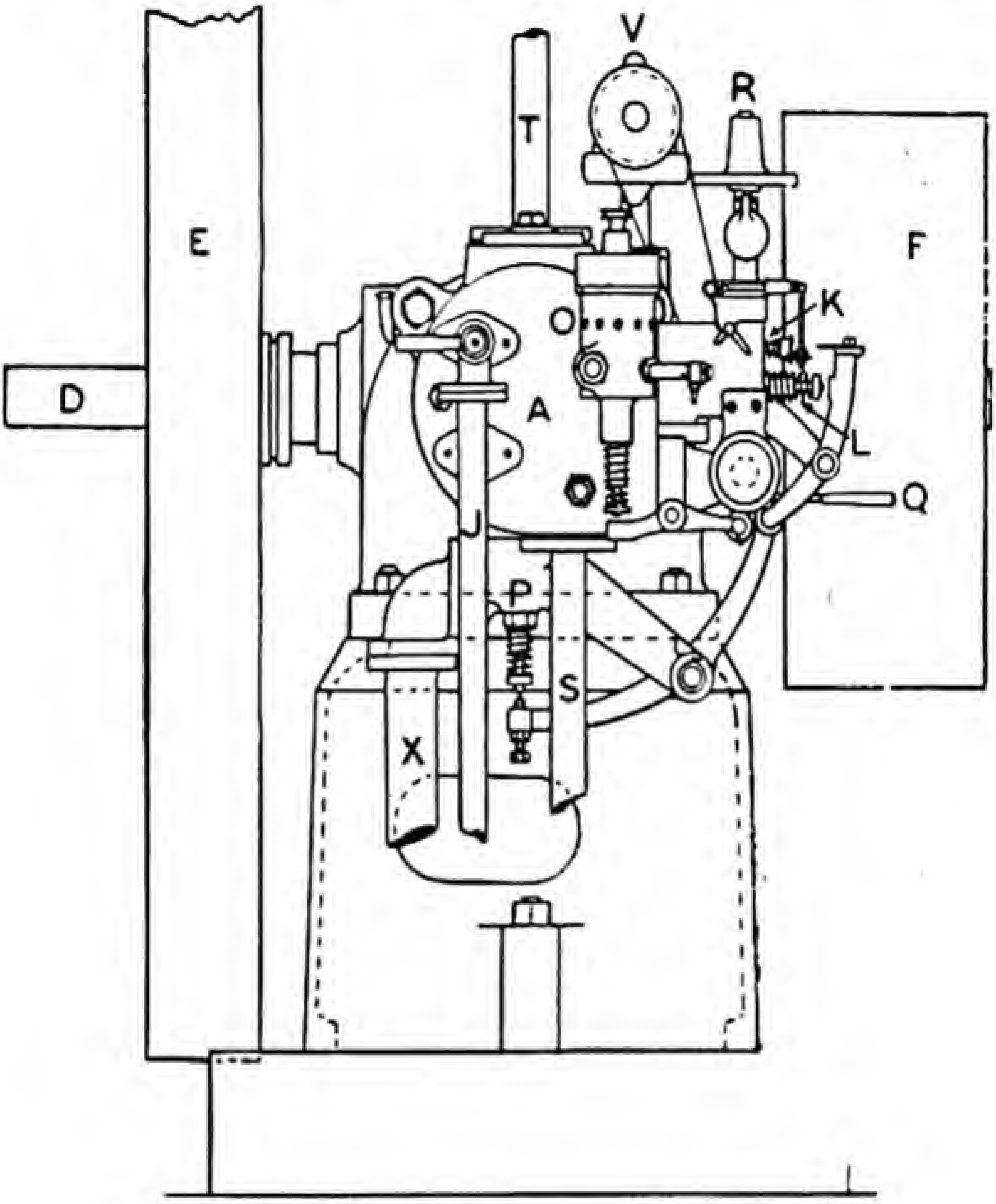


FIG. 305.—End elevation of 6½ B.H.P. Crossley gas engine.

set screws. The construction of the connecting rod will be understood by reference to Fig. 306.

D is the crank-shaft, cut out of the solid and carrying a flywheel *E* and a belt pulley *F*. A balance weight is cast with the flywheel and is placed opposite the crank. *G* is the side shaft, driven from the crank shaft by screw wheels *H*. The side shaft makes one revolution for two of the crank shaft and carries at its left-hand end the various cams for operating the valves and a bevel wheel for rotating the centrifugal governor *R*.

Gas is supplied to the engine through a pipe *J*. The air supply is taken from the interior of the soleplate through an air silencer

N. *K* is the gas admission valve and *L* is a valve through which the mixed gas and air pass into the cylinder. *O* is the hot tube ignition arrangement. The exhaust valve is at *P*, and the exhaust

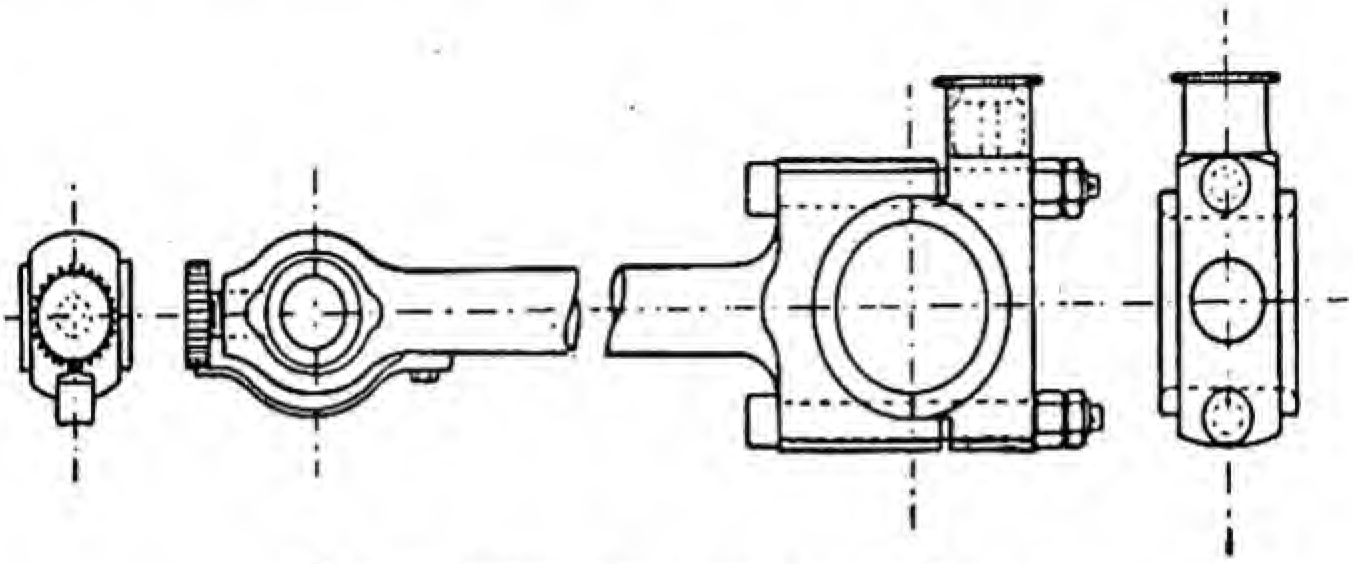


FIG. 306.—Crossley gas engine connecting rod.

gases are discharged to the atmosphere through a pipe *X*. *Q* is a handle used in starting the engine, and by its use, as will be explained, later ignition of the charge and lower compression are obtained. *V* is a lubricator supplying oil to the engine.

The supplying of the charge.—This will be better understood by reference to Fig. 307. The admission valves are contained in a casting bolted to the side of the cylinder, which also serves to support the governor and the lever for operating the valves. Gas is led from the gas supply pipe along the passage *A* formed in the cylinder casting and enters the valve chamber. Here it will pass the gas admission valve *B* if open, and will enter a mixing chamber *G*. Air is brought from the air silencer through the passage *F*, formed in the cylinder casting, and is mixed with the incoming gas in the chamber *G*. The mixed charge finally enters the cylinder past the admission valve *E*.

The valves are held to their seats by external springs, and are pushed open at the proper instants by a lever *C* which is operated by a cam *D* on the side shaft. A small roller mounted on the end of the lever bears on the edge of the cam. The admission valve *E* is pushed open every charging stroke, so that a supply of air is always taken in. The gas admission valve, however, may not be operated every charging stroke as it is under the control of the governor. The governor, which is of the dead-loaded pendulum type, will raise the lever *H* should the speed of the engine be too

high. Attached to *H* is a hanging rod *K*, guided at its lower end by a slot cut in the plate *L*, which works in a collar on the valve stem. To the end of the lever *C* is attached a thin steel plate *M*;

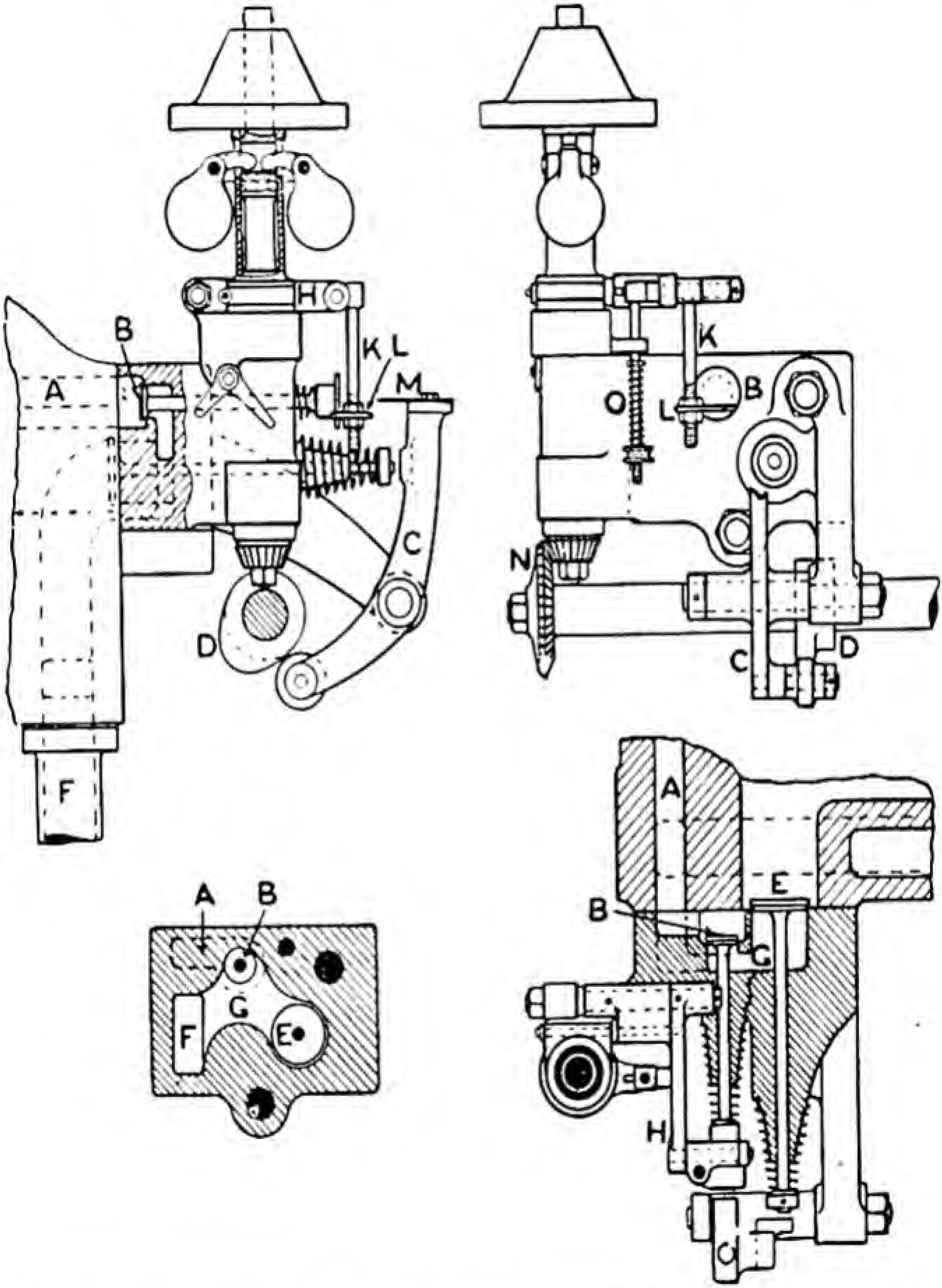


FIG. 307.—Admission and governing arrangements of the Crossley gas engine.

the edge of the plate *L* which is nearer to *M* has a V notch cut in it. Should the plate *L* be exactly opposite *M* when the lever *C* is operated by the cam, the plate *M* will push the valve stem by aid of the intervening plate *L*, the sharpened edge of *M* engaging with

the V notch in the edge of L, and thus the gas admission valve will be opened and the gas supply will be drawn into the cylinder. But should the governor, by reason of the engine speed being too high, have drawn the plate L above the level of M, the plate M will miss L when the lever is operated, and no gas will be supplied. In this latter case, no explosion will follow and consequently the speed of the engine will be lowered. This method is called governing by **hit or miss**; engines worked on this system receive at each charging stroke a charge of full strength or of no strength, *i.e.* air alone.

In addition to the conical dead load on the top of the governor spindle, there is a spring O by means of which a small additional load may be applied. The spring is regulated by lock nuts on the spring spindle, and by use of these the engine speed may be varied to a small extent. The governor is driven from the side shaft by the bevel wheels N.

Ignition of the charge.—The hot tube ignition arrangement is shown in Fig. 308. As metal tubes, when at a high temperature, are more or less affected by the action of the gases, a porcelain tube is used in this ignition arrangement. A is the porcelain tube contained in a cast-iron case B fitted with a removable cover C, which projects downwards into B so as to form a double-walled case round the tube. The inner case is lined with asbestos so as to assist, by keeping down conduction of heat, in maintaining the temperature of the tube. The tube is held in position by means of a screw D fitting a tapped hole in the cover. A metal cap E is placed between the screw and the top of the tube so as to close its upper end. As the porcelain of the tube expands considerably on heating, provision must be made for allowing this to occur in the direction of the length of the tube. This result is secured, and a gas-tight joint made, by means of an asbestos washer in the socket at the lower end of the tube and another between its upper end and the cap E. The asbestos possesses just enough springiness to accommodate the expansion.

The tube is heated by means of a ring of gas jets supplied with mixed gas and air by means of the Bunsen tube F. Additional air is drawn in through air holes G, and this air, circulating between the two casings B and C, serves to keep the outer case cool.

The interior of the porcelain tube is put into communication alternately with the air supply passage and, when ignition is required, with the interior of the cylinder. *H* is the passage through which the charge enters the cylinder after passing the admission valve. *J* is the passage through which the air supply is brought to the valve case. An ignition valve *K*, when in the position shown, permits free communication between the porcelain tube and the contents of the cylinder. *K* is held down by means of an external spring, and is pushed upwards when required by a

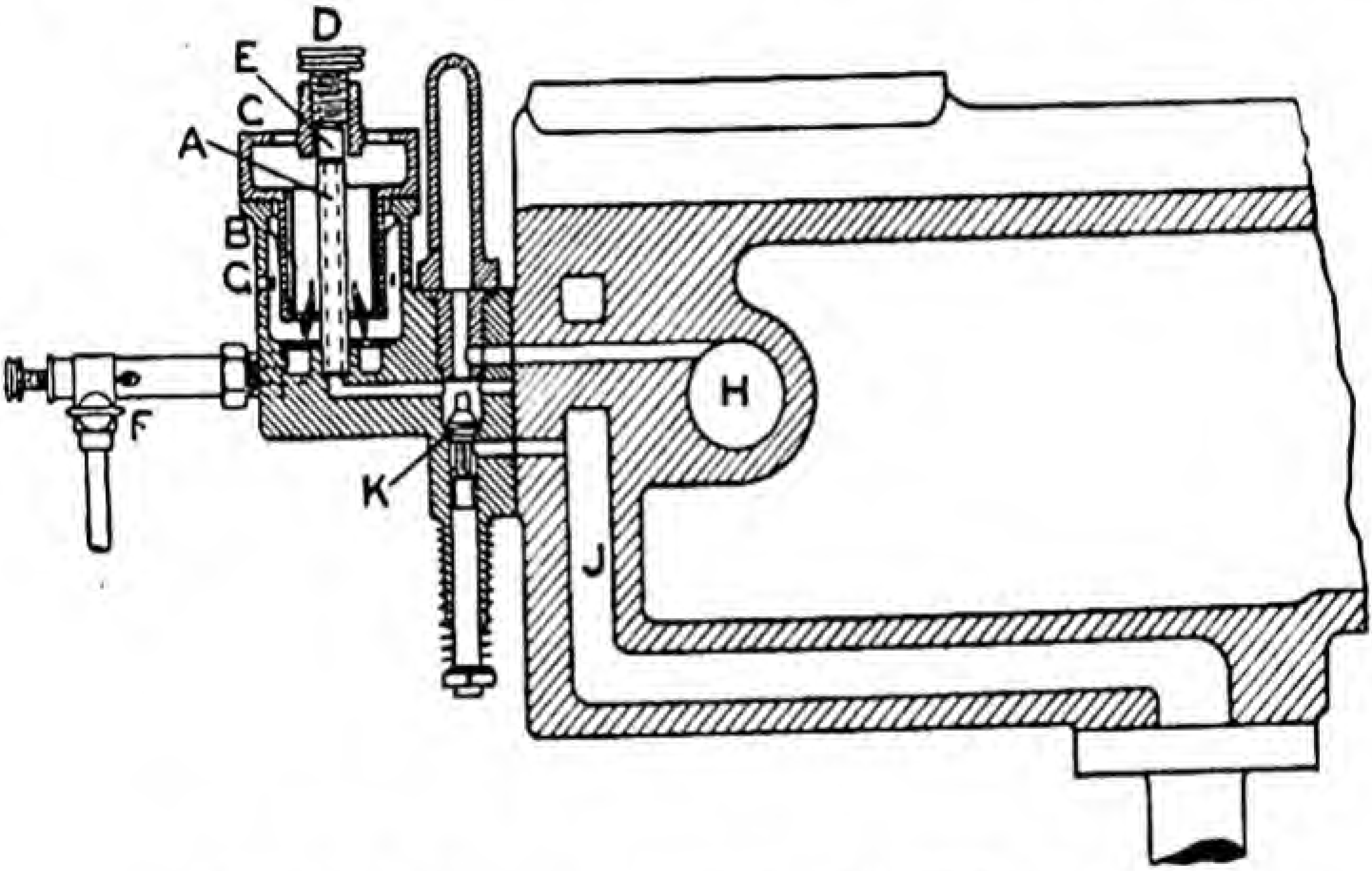


FIG. 308.—Ignition arrangements in the Crossley gas engine.

lever operated by a cam on the side shaft. When raised, the small cylindrical projection on its upper end enters and closes the passage between the porcelain tube and the cylinder.

The action is as follows: During the charging and compression strokes, the ignition valve *K* is pushed upwards and so keeps the ignition tube in communication with the atmosphere and out of communication with the cylinder. The pressure in its interior will therefore be equal to that of the atmosphere. This process is called ventilating the tube. Towards the end of the compression stroke, when the pressure of the charge in *H* is approaching its maximum value, the ignition valve *K* is dropped suddenly by the action of the cam, thus opening communication between the cylinder and the hot tube and closing the opening to the atmosphere. A

portion of the charge rushes into the hot tube and ignites on coming into contact with its walls. At once its pressure greatly increases, and the flame is shot back along the passages into the port *H* and so into the heart of the charge in the cylinder, thus igniting the whole.

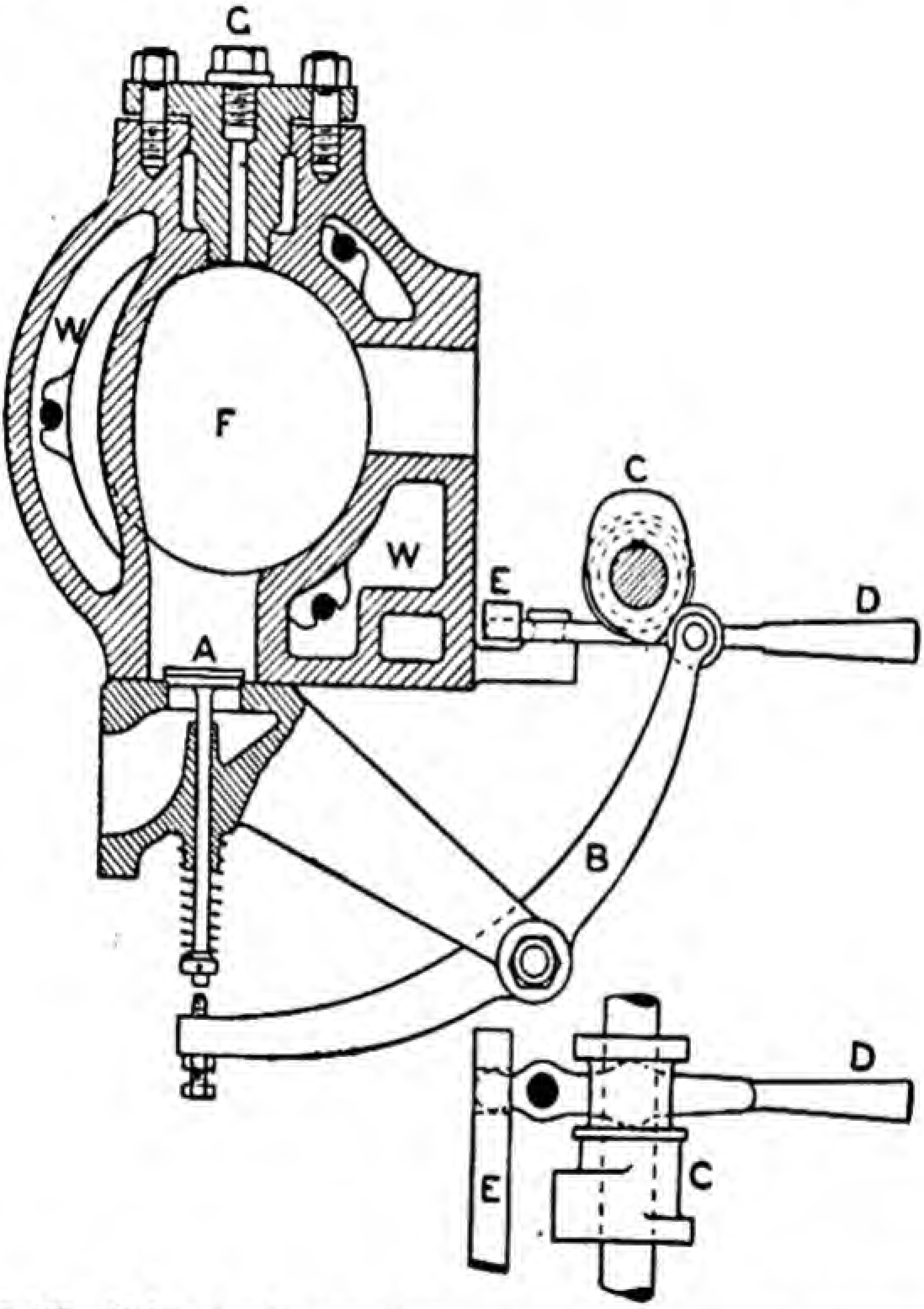


FIG. 309.—Exhaust valve and starting gear of the Crossley gas engine.

The exhaust.—The exhaust arrangements are shown in Fig. 309. *A* is the exhaust valve, placed in a casting bolted to the lower side of the cylinder *F*. The valve is held to its seat by an external spring, and is operated by a lever *B* and cam *C* on the side shaft. The cam *C* is two-stepped, and can be moved axially on the side shaft by the handle *D*. When in running position, the cam is so placed that the larger projection on its rim alone operates the

lever, and this will occur during the exhaust stroke. When starting the engine, the cam is moved along to the position in which both projections operate the lever. In this position the exhaust valve will be opened twice each cycle, viz. during the exhaust stroke and also during the early part of the compression stroke, the smaller projection on the cam effecting the latter operation. The result is that compression does not begin until later in the stroke, and a very much lower compression pressure is reached. During starting the engine has to be turned by hand for a few revolutions while the first charge is drawn in, compressed

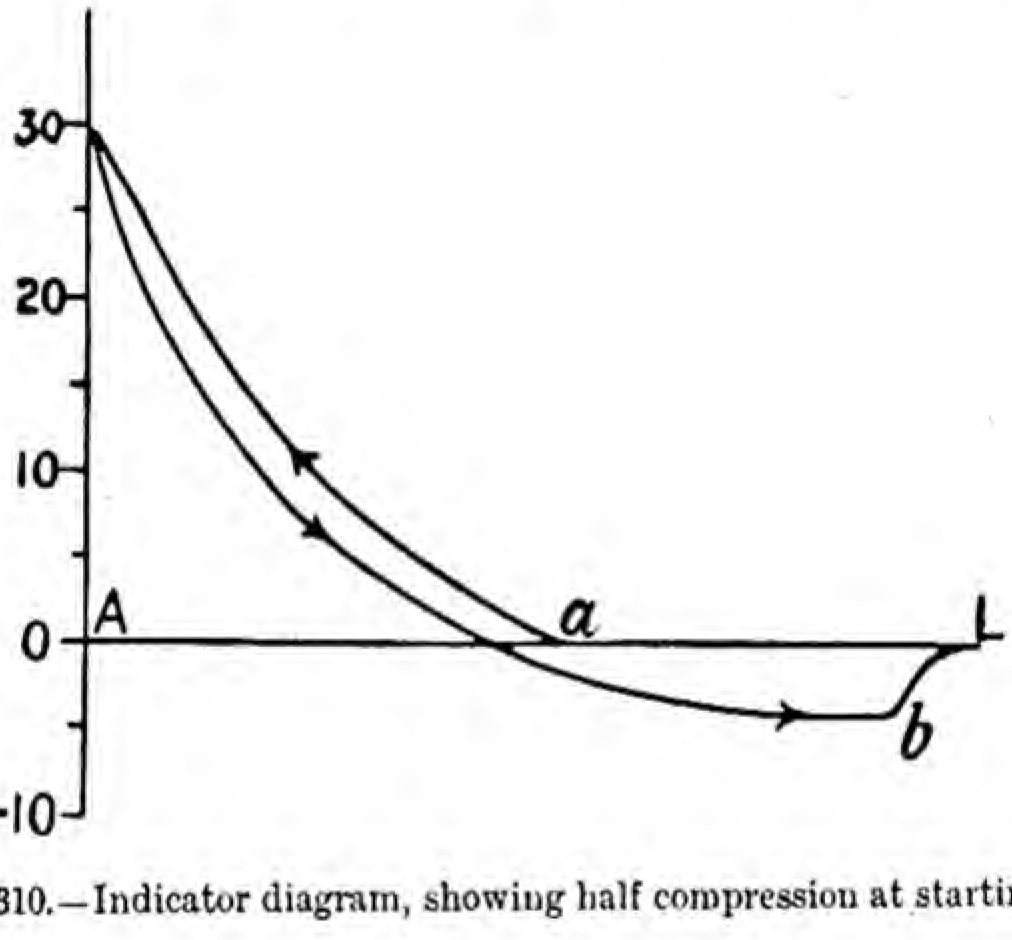


FIG. 310.—Indicator diagram, showing half compression at starting.

and ignited, and it would not be easy to do this if full compression were given.

Further, ignition of the charge, when the engine is working at its normal speed, begins a little before the end of the compression stroke when the crank has not yet reached the dead centre. Should this setting of the ignition valve be used when starting, while the engine is being slowly turned by hand, there is risk that explosion will occur before the crank reaches the dead centre. In this case the engine will start off suddenly in the wrong direction of rotation and the attendant may be injured. To obviate this risk, the cam operating the ignition valve is two-stepped. One step gives ordinary working ignition, the other is so set as to produce ignition after the crank is over the dead centre. The rod

E (Fig. 309), operated by the same movement of the starting handle which puts the double-projectioned exhaust cam into gear,

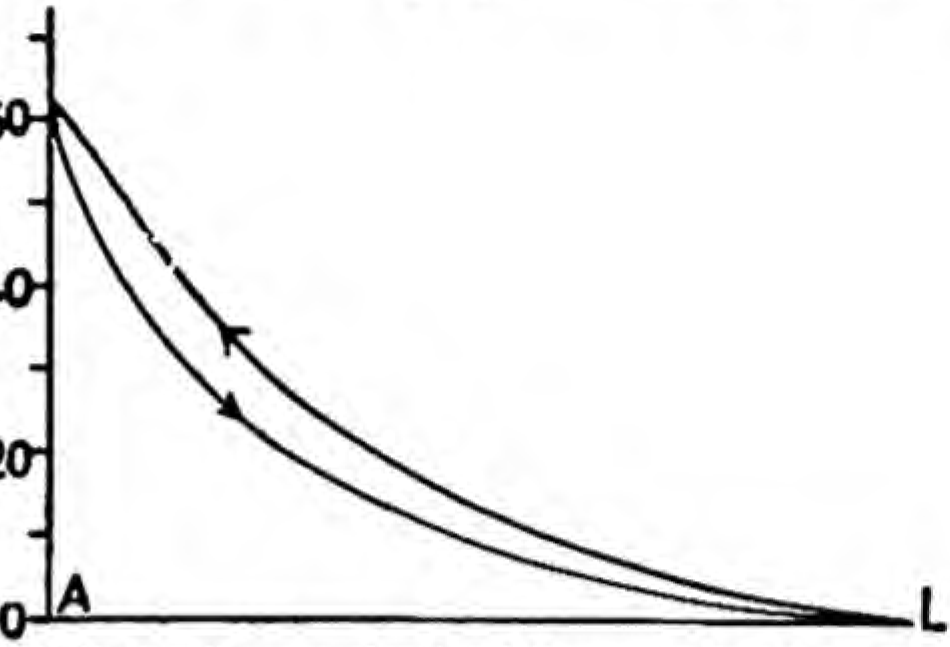


FIG. 311.—Indicator diagram, showing full compression during working.

is connected to the ignition valve lever, and brings it into gear with the late ignition cam. One movement of the handle *D* therefore secures low compression and late ignition at starting.

To show the difference in compression during starting and working, the indicator diagrams reproduced in Figs. 310 and 311 were taken while the engine was turned by hand as rapidly as possible. No gas was admitted. Fig. 310 shows the low compression used in starting the engine; the maximum pressure attained was $28\frac{1}{2}$ lbs. per square inch above atmospheric pressure. The exhaust valve closes at *a* on the compression stroke, and during the succeeding stroke the compressed air expands again, falling below atmospheric pressure. The exhaust valve re-opens at *b*. In Fig. 311 is seen full compression. The maximum pressure on this diagram is $62\frac{1}{2}$ lbs. per square inch above atmospheric pressure.

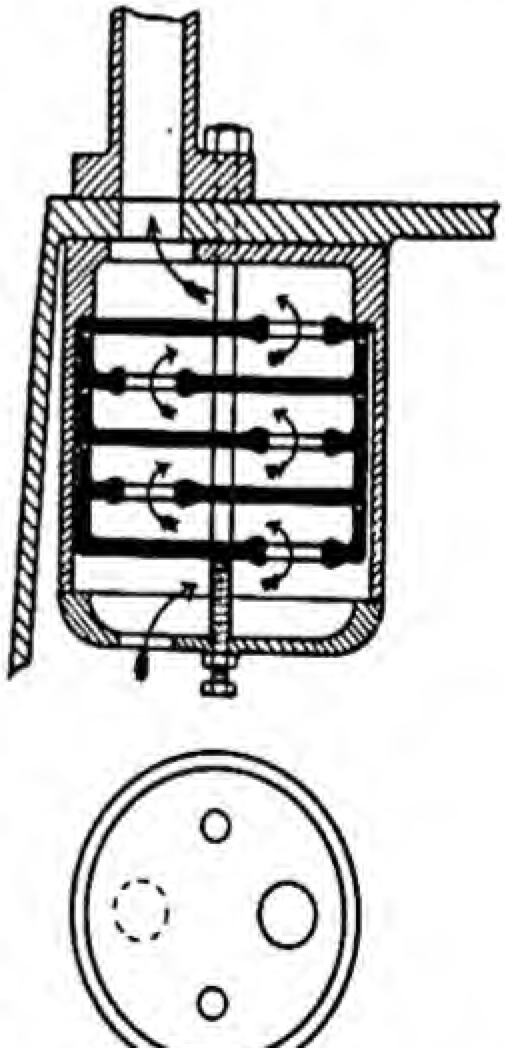


FIG. 312—Crossley air-silencing box.

Air supply to the cylinder.—The supply of air to the cylinder is drawn from the interior of the soleplate. In order to reduce the noise caused by the suction, an air silencer (Fig. 312) is fitted.

This consists of a cylindrical box containing a number of circular plates, each plate having an air hole placed eccentrically. The plates are so disposed in the box that the air has to travel along a sinuous path as shown by the arrows.

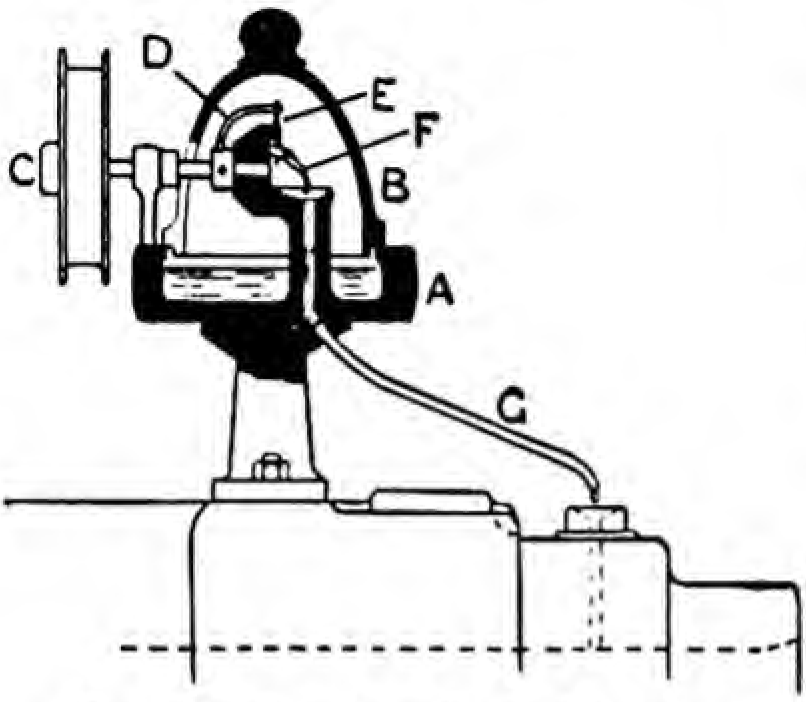


FIG. 313.—Crossley cylinder lubricator.

Cylinder lubricator.—A regular supply of oil is maintained to the cylinder by means of the lubricator shown in section in Fig. 313. *A* is a wide, shallow cup containing oil and closed by a loose cover *B*. A small spindle *C* rotates within the cover and is driven by a belt from the side shaft. *D* is a crank secured to the spindle and having a piece of wire *E* loosely

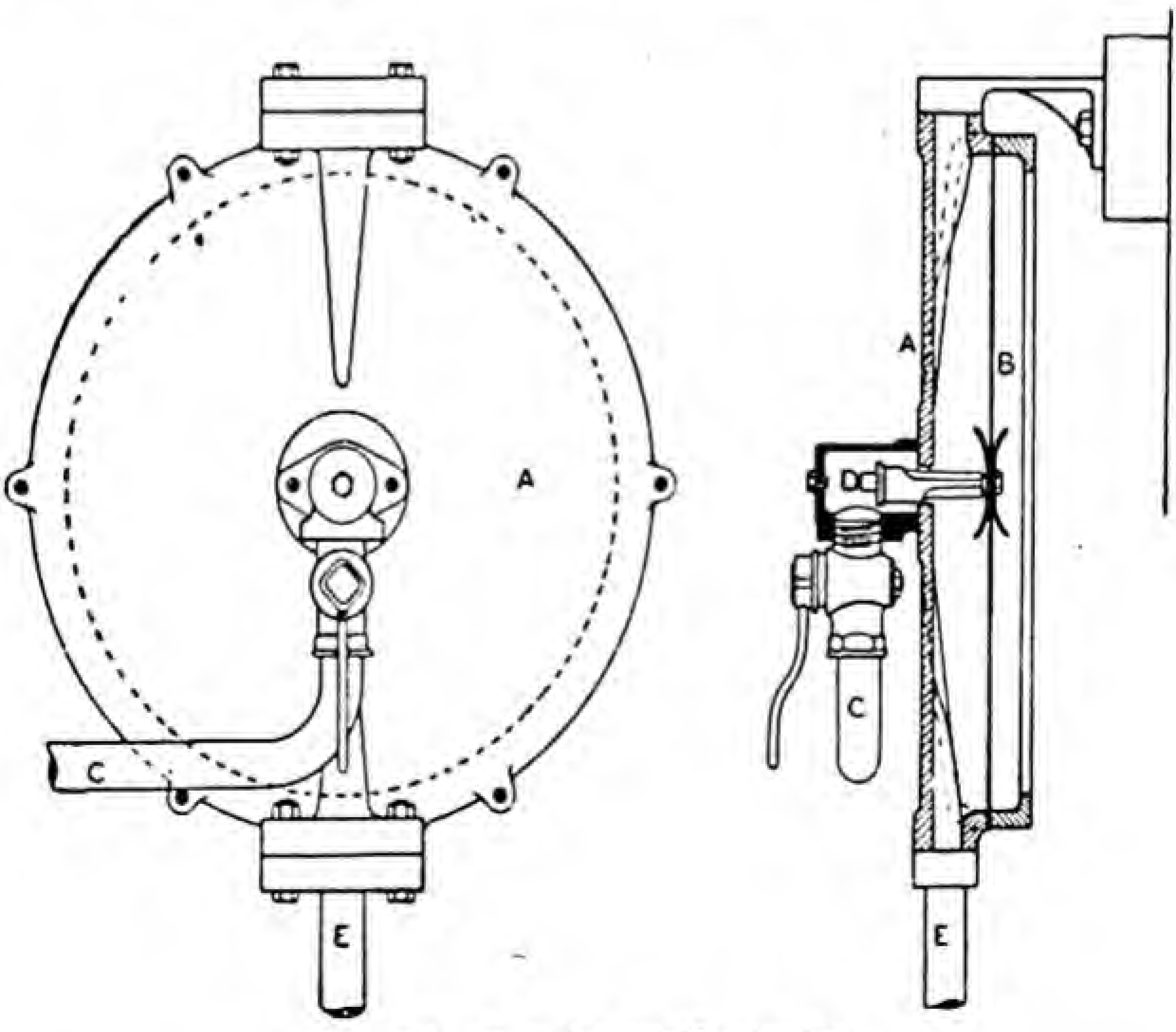


FIG. 314.—Crossley antifluctuator.

attached to it. On the spindle rotating, the crank will come to the bottom, and the wire *E* will dip into the oil and pick up a drop.

On the crank rising again, the drop of oil is wiped off the wire E by another fixed wire F. The drop of oil then falls from the end of F into a pipe G leading to the cylinder. Notice that this arrangement will keep up a supply of oil while the engine is running, and automatically ceases to act when the engine is stopped.

Antifluctuator.—This is a device placed in the gas supply pipe between the cylinder and the gas-meter, and has for its object the prevention of the rush of gas into the cylinder during the charging stroke affecting the meter. It consists of a circular cast-iron box A (Fig. 314) closed at the back by an elastic rubber diaphragm B. Gas enters the box from the meter through the pipe C and valve D, and by its pressure pushes the diaphragm outwards. The valve D is attached to the diaphragm, and so gradually closes as the box fills. E is the supply pipe to the engine. When the charging stroke occurs, the engine draws its supply of gas from the antifluctuator at a time when, as the box is full of gas, the valve D has cut off communication with the meter. Immediately the charging stroke is over, the elastic diaphragm re-opens D and the box again fills with gas.

EXERCISES ON CHAPTER XIX.

1. Explain the four stroke or Beau-de-Rochas cycle. Sketch an indicator diagram and name each stroke on it.

2. Sketch a gas engine cylinder in section, omitting the piston. Show the valves with as little detail as possible. Name the parts and explain what each is intended for.

3. Sketch and describe the piston and connecting rod of a gas engine.

4. Describe with sketches the method of driving the valves of a gas engine. What has to be done at starting? Give reasons.

5. Give sketches and description of a gas engine governor. What is the reason for the use of the hit-or-miss action in small engines?

6. Explain and give sketches of any device for igniting the charge in a gas engine cylinder.

7. Why are heavier flywheels required for gas engines than for steam engines of the same power?

8. The piston of a gas engine is 6·7 inches in diameter by 1·2 feet stroke. At 200 revolutions per minute there are 80 explosions per minute and the engine gives 6½ I.H.P. Calculate what the mean pressure must have been.

9. The crank shaft of a gas engine is giving out steadily 20 horse-power at an average speed of 150 revolutions per minute. How many foot-pounds is this per cycle (of two revolutions)? About how much of this must be stored and unstored by the flywheel if there are 75 explosions per minute? 1904.

CHAPTER XX.

OIL ENGINES.

Cycle in oil engines.—Most oil engines in use at the present time follow the Beau-de-Rochas cycle of operations, the liquid fuel being vaporised and mixed with air so as to form an **explosive mixture**. The mixture is drawn into the cylinder and used in precisely the same manner as in the gas engine previously described (p. 375). One important modification of this cycle is to be found in the **Diesel oil engine**. In this engine air only is drawn in during the suction stroke. Compression of the air to a high pressure (about 500 lbs. per sq. inch) follows, with the result that the temperature rises to about 1000° F. The next out-stroke of the piston is the working stroke. During the early part of this stroke, the oil fuel is sprayed into the cylinder and is ignited on coming into contact with the hot air contained therein. The supply is cut off after a small fraction of the stroke has been completed, and expansion follows. Exhaust is accomplished during the next in-stroke of the piston. The importance of this cycle from the theoretical point of view lies in (*a*) the approximately adiabatic compression of the air to a temperature at which the fuel will ignite, (*b*) slow combustion while the piston is moving out, allowing expansion of a practically isothermal nature to take place, (*c*) approximately adiabatic expansion of the products after the point of cut-off. This engine shows the highest heat efficiency attained up to the present; in a test on a 35 B.H.P. Diesel engine at Harrogate on May 10th, 1902, Mr. H. Ade Clark, found that 36·3 per cent. of the heat supplied in the oil was converted into energy given to the piston, and 28·7 per cent. of the heat supplied in the oil was converted into useful work, as

shown by the Brake Horse Power. The engine uses less than 0·5 lb. of oil per B.H.P. per hour when working at nearly full power, and about 0·8 lb. when working at light loads.

Vaporising the oil.—In engines using the Beau-de-Rochas cycle, the oil is prepared for introduction into the cylinder in several different ways. In many engines, such as the Crossley and the Tangye oil engines, the oil is sprayed into a vaporising chamber which is heated externally by a lamp kept burning continually while the engine is running. In the vaporiser, the spray is mixed with air which has been previously warmed, and is converted into a vapour ready to be swept into the cylinder during the charging stroke. In other forms of vaporiser, such as the Priestman, the vaporiser is kept warm by a jacket through which the exhaust gases pass on their way from the cylinder to the atmosphere. In the Hornsby-Akroyd oil engine, a chamber at the rear of the cylinder, and in constant communication with it, is kept hot by the heat generated during the combustion in the cylinder. The chamber has no water jacket, and, when the engine is loaded so that a large quantity of heat is being generated in the cylinder, becomes red hot. Oil is sprayed direct into it, and the resulting vapour is mixed with the air supply drawn into the cylinder through a separate port. All these devices are suitable for the successful treatment of **heavy oils**, such as Royal Daylight (American) and Russolene (Russian) oils. The oil is supplied under a head obtained either by placing the oil supply tank at an elevation whence the oil gravitates to the vaporiser, or by the use of a pump driven by the engine.

In engines using **light oils**, such as petrol, the vapour is much easier to obtain. These light oils give off inflammable vapour even at ordinary atmospheric temperatures, and consequently are very easily vaporised. It is sufficient merely to spray the oil into the incoming supply of air. The vaporiser, or carburettor as it is called in such cases, is placed close to the engine cylinder, and so is slightly warmed by conduction of heat from the cylinder. The effect is to produce carburetted air, *i.e.* air charged with petrol vapour, forming an explosive mixture.

Action in an oil engine.—To enable the action of and arrangements in an oil engine to be understood, the Hornsby-Akroyd oil engine has been selected for description. The general appearance

of the engine is shown in Fig. 315, and sectional elevations are given in Figs. 316 and 317. Referring to the latter illustrations, *A* is the cylinder, open at the left-hand end, and surrounded by a water jacket supplied with cooling water through a pipe *K*; the water is discharged through the pipe *L*. *B* is the piston to which one end of the connecting rod is attached, the other end being connected to the crank. The crank shaft carries a heavy flywheel, and drives, by means of screw wheels, a side shaft on which are mounted cams for operating the valves, and a bevel wheel for driving the centrifugal governor *M*.

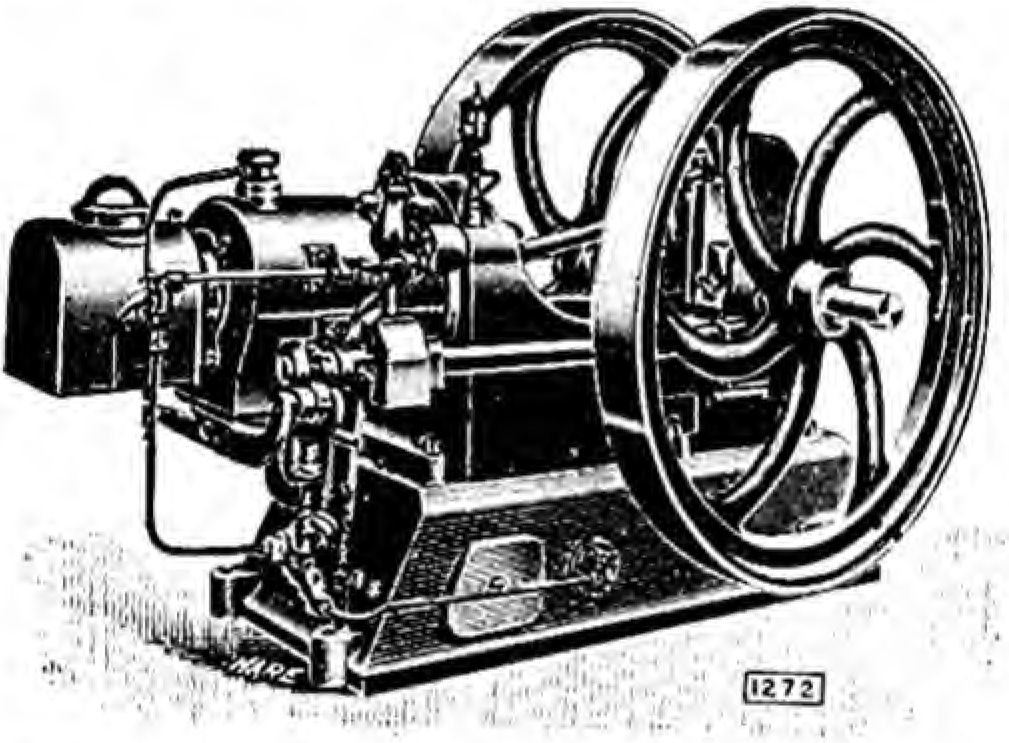


FIG. 315.—External view of the Hornsby-Akroyd oil engine.

C is the vaporiser, which is left unjacketed at its rear portion. This portion is kept hot while the engine is running by heat from the combustion of the fuel in the cylinder, and is brought to a suitable temperature before starting the engine by means of an external lamp *D*. After the engine is started, this lamp is put out. The oil supply to the cylinder is drawn from the tank *E* formed in the bed plate. The oil passes through a strainer in the tank, which removes grit, then through a pipe *F* to a pump *G*. The plunger of the pump *G* is operated, as will be seen in Fig. 317, by the same lever which opens the air intake valve, and, on the down stroke, forces a small quantity of oil through a pipe *H* (Fig. 316) into the vaporiser *C*. Thus oil is supplied to the cylinder and the air valve is opened to take in the requisite air during the same out-stroke of the piston.

The air and exhaust valves are placed in a chamber *N* (Fig.

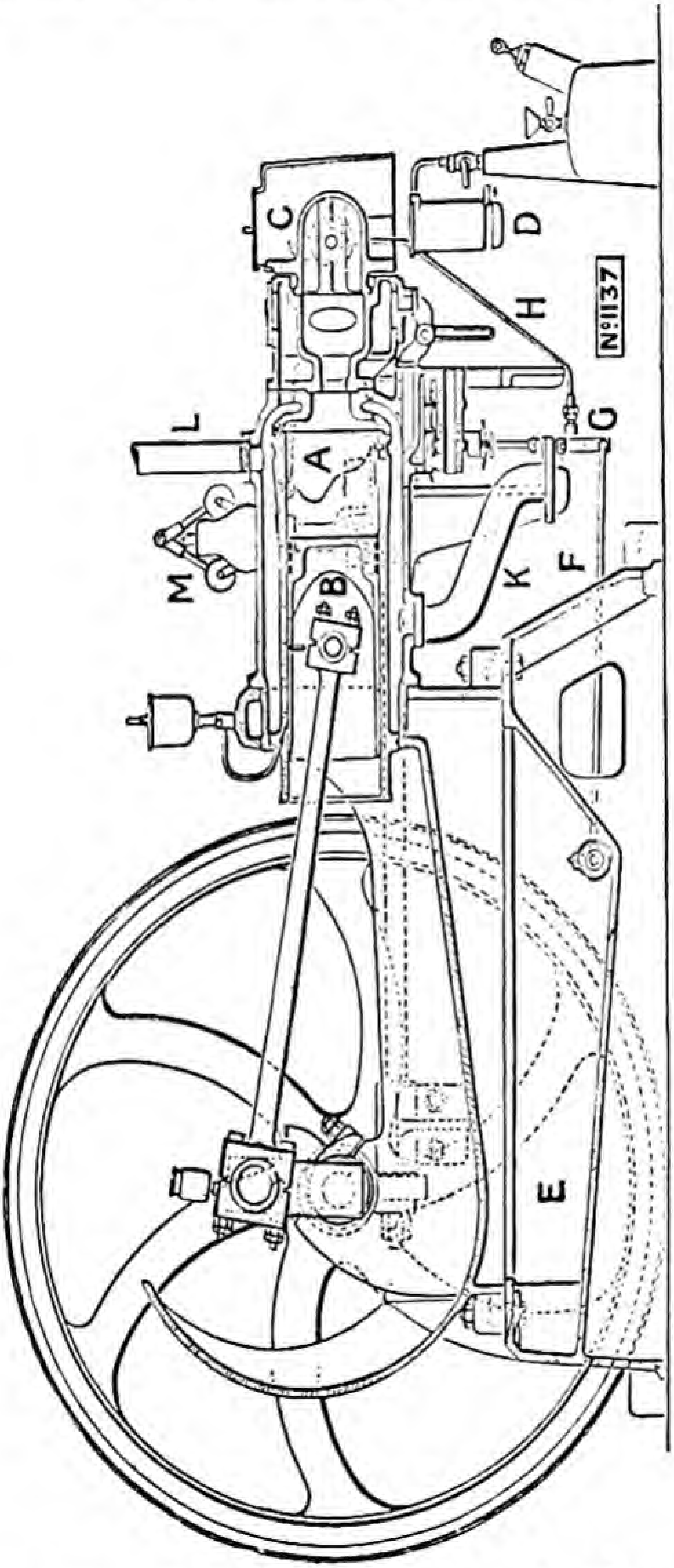


Fig. 316.—Sectional side elevation of the Hornsby-Akroyd oil engine.

317), which communicates with the cylinder through a port *O*. Each valve is operated at the required instants by a lever worked

by a cam on the side shaft. *P* and *Q* are the lever and cam for working the air valve and the oil pump. The air valve always opens and closes at the same instant. To enable the engine to be turned easily by hand when starting, the exhaust valve cam is

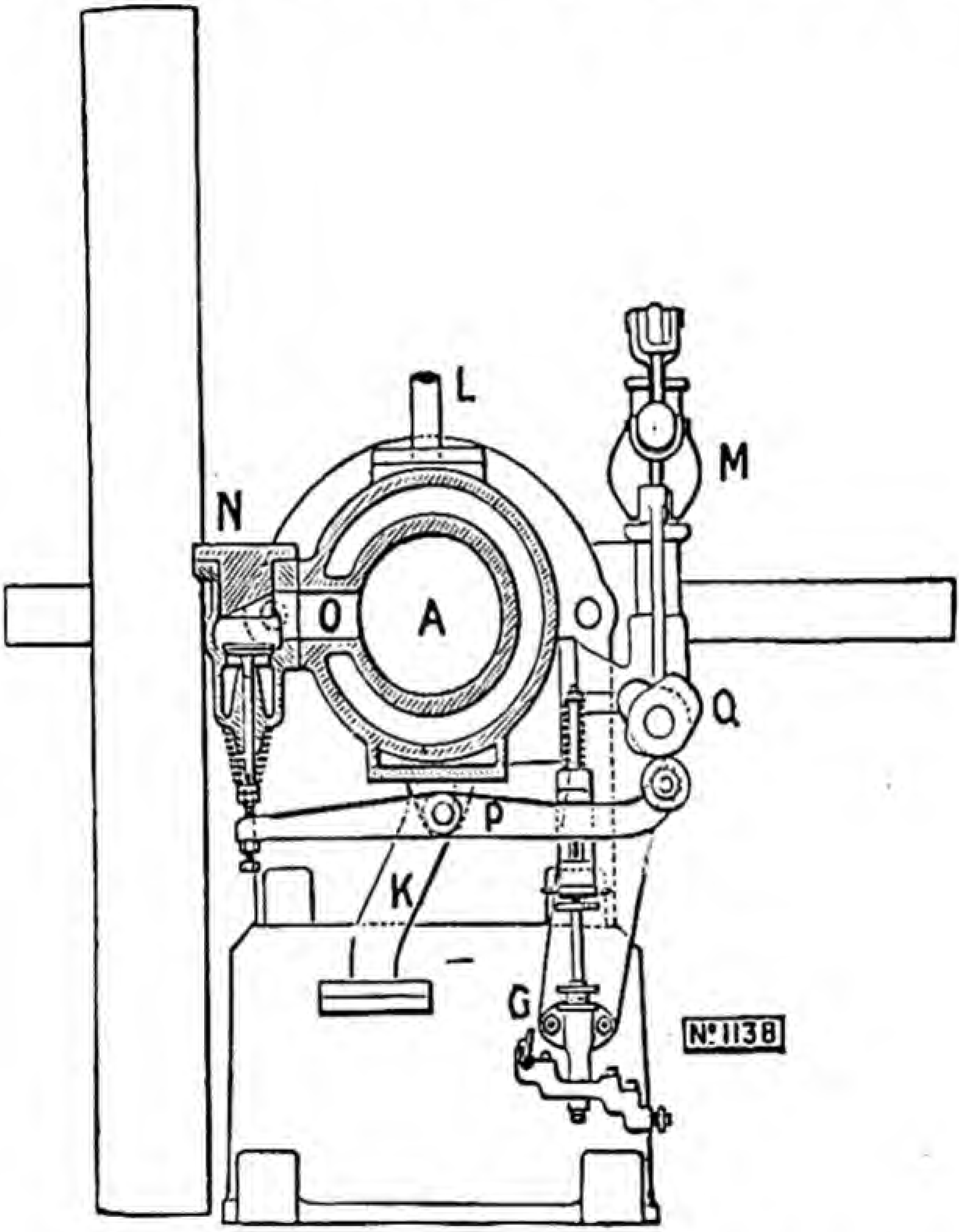


FIG. 317.—Sectional end elevation of the Hornsby-Akroyd oil engine.

two-stepped, and either step may be used to operate the valve. The starting step has a projection which opens the exhaust valve during the early portion of the compression stroke, and so reduces the compression pressure in the manner described for the Crossley gas engine (p. 385).

Ignition and governing.—Fig. 318 shows the vaporiser valve box in section, mounted in position on the side of the vaporiser.

Oil under pressure from the pump forces open the horizontal valve and passes into the vaporiser through a very fine hole. The overflow valve at the top of the box is held up against its seat by means of an external spring, and, should the engine speed be too high, will be pushed open by the action of the governor communicated through rods and levers.

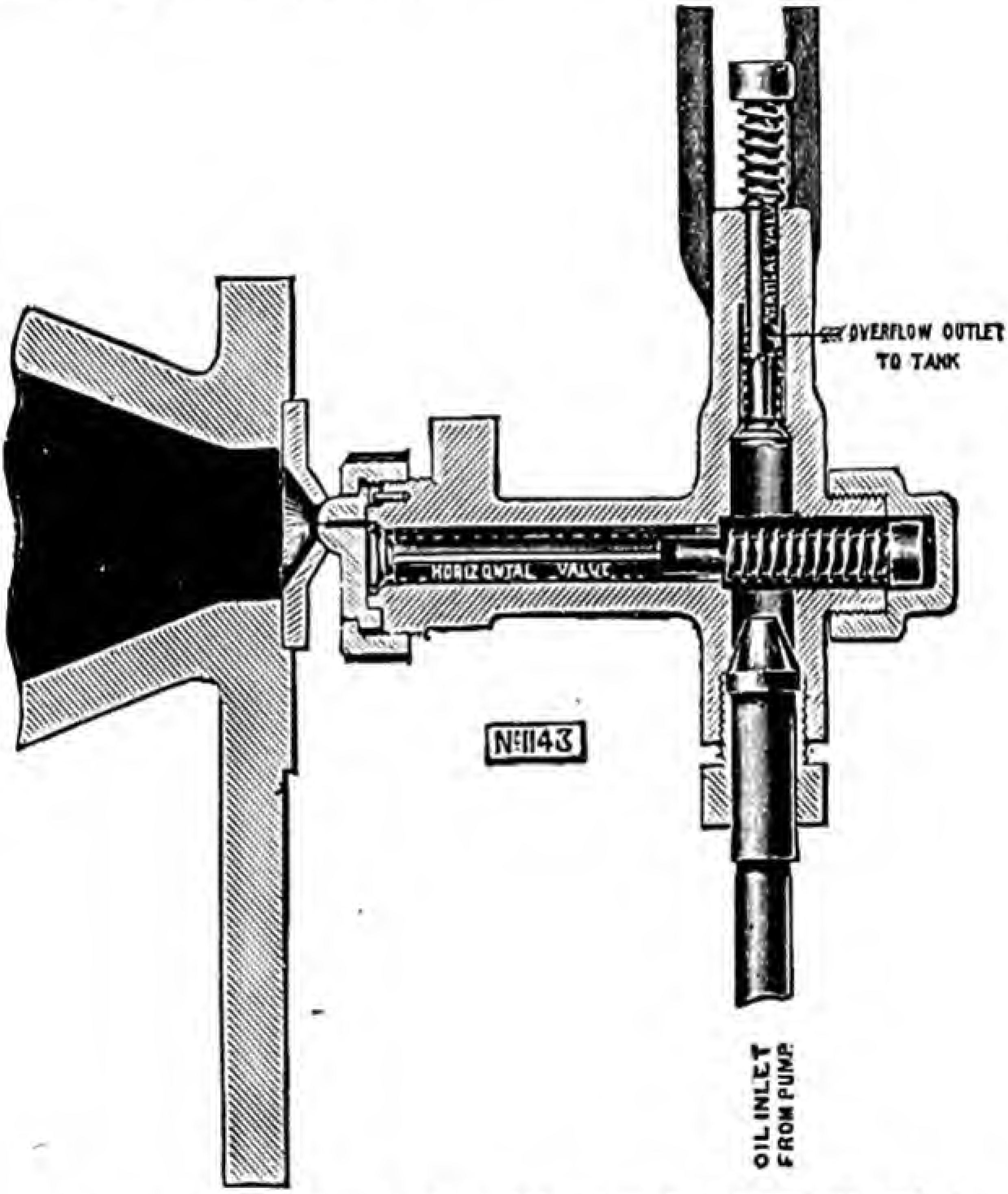


FIG. 318.—Vaporiser valve box of the Hornsby-Akroyd oil engine.

When open, the overflow valve permits the oil supplied by the pump, or part of it, to escape through an overflow outlet, and so return to the oil supply tank instead of finding its way into the cylinder. The supply of fuel to the cylinder is thus regulated to suit the work which the engine is called upon to perform, and the speed is maintained steady.

Ignition of the charge is secured by the action of the hot walls of the vaporiser. The amount of compression is adjusted carefully to

suit the class of oil being used, and, when so adjusted, explosion takes place immediately on the end of the compression stroke being reached, and just as expansion begins. The regularity of firing in this engine is most remarkable, when it is considered that an explosive mixture is in contact with hot walls during the compression stroke, but it is found that ignition will not occur (in a properly adjusted engine) until expansion is just about to commence.

Petrol motors.—Small motors using petrol (petroleum spirit) are used extensively for operating motor cars and cycles. The B.H.P. obtainable from one cylinder does not exceed 9 or 10, and usually the engines are smaller. Cycle motors range from 2 to 4 B.H.P. To obtain a larger power for motor car work the number of cylinders is increased up to as many as six or eight. The arrangement is convenient, for the cranks can be placed relative to one another in such a manner as to produce a balanced or nearly balanced machine. Motor car engine cylinders are generally water-cooled, cycle engine cylinders are air-cooled. The cycle usually followed is the ordinary Beau-de-Rochas. Ignition is generally secured by means of an electric spark.

Description of cycle motor.—In Fig. 319 a small cycle motor is shown in section and in end elevation. *A* is the cylinder, of cast-iron, fitted with a cast-iron piston *B*, which is kept gas-tight by means of spring rings. The cylinder is kept cool by the device of exposing as large a surface as possible to the currents of air flowing past the machine when in motion. To secure this, a large number of thin ribs project from the outer surface of the cylinder. The connecting rod *C* is a mild steel stamping, and is connected to the piston by a pin *D*, and at the lower end to the crank pin *E*. *FF* are flywheels, secured by nuts and keys to the crank shaft *GG* and to the crank pin *E*. *H* is a shaft carrying two cams *O* and *S*, used respectively for operating the exhaust valve and the ignition arrangement. *H* is driven at half the speed of revolution of the crank shaft by means of toothed wheels. The flywheels, gearing, etc., are contained in a crank-chamber *J*, which, for the sake of lightness, is constructed of aluminium. The cylinder *A* is secured to the crank chamber by means of studs. The explosive mixture enters the cylinder during the suction stroke through the pipe *K* and admission valve *L*. In the motor illustrated, the admission

valve is not mechanically operated, but is simply drawn open during the suction stroke by the rush of gas entering the cylinder. At other times it is held to its seat by a light spring. In later patterns of small motors the admission valve is operated at the

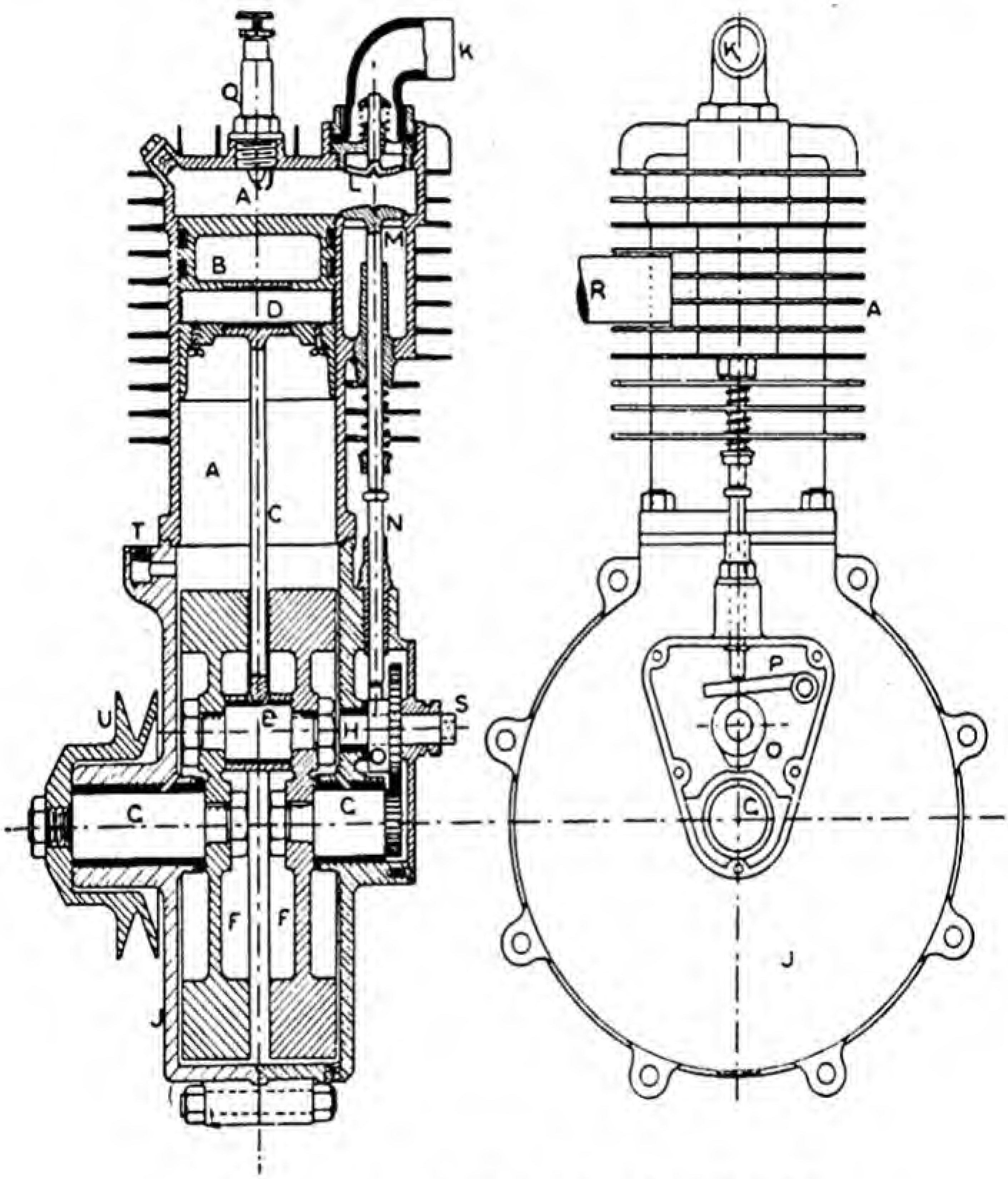


FIG. 319.—Section and side elevation of a cycle motor.

right instant by means of a cam on the shaft *H* and rods leading from it to the valve. *M* is the exhaust valve, held down on its seat by an external spring and operated from the cam *O* through a small lever *P* and a tappet rod *N*. Ignition of the explosive mixture is obtained by use of an electric spark, the current being supplied from a small accumulator, which, of course, must be

charged periodically. A coil, consisting of two parts, viz. an inner primary coil through which the accumulator current passes, and an outer secondary coil having a large number of turns of much finer wire insulated from the primary coil, is used to increase the potential of the current and to enable a spark to cross between two platinum points having a small gap between. These points are attached to an ignition plug *Q*, and are situated, as shown, inside the cylinder *A* near to the admission port. The spark will therefore pass in a space containing rich explosive mixture.

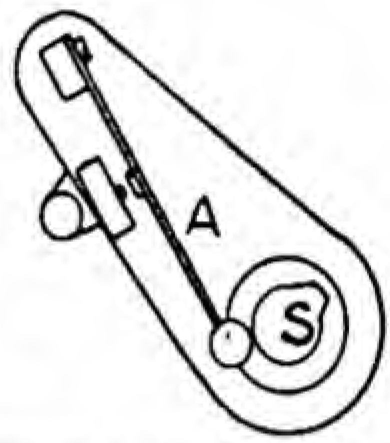


FIG. 320.—Cycle motor contact maker.

The spark is timed by means of a cam *S* (Fig. 320) mounted on the cam shaft. The projection on the cam pushes a spring *A* at the proper instant and causes it to come into contact with a fixed platinum point, thus closing the circuit. *U* is an external pulley having a grooved rim (Fig. 319); a band passes from this to a similar but larger pulley on the rear wheel of the cycle, and so drives the wheel at a speed of rotation lower than that of the motor.

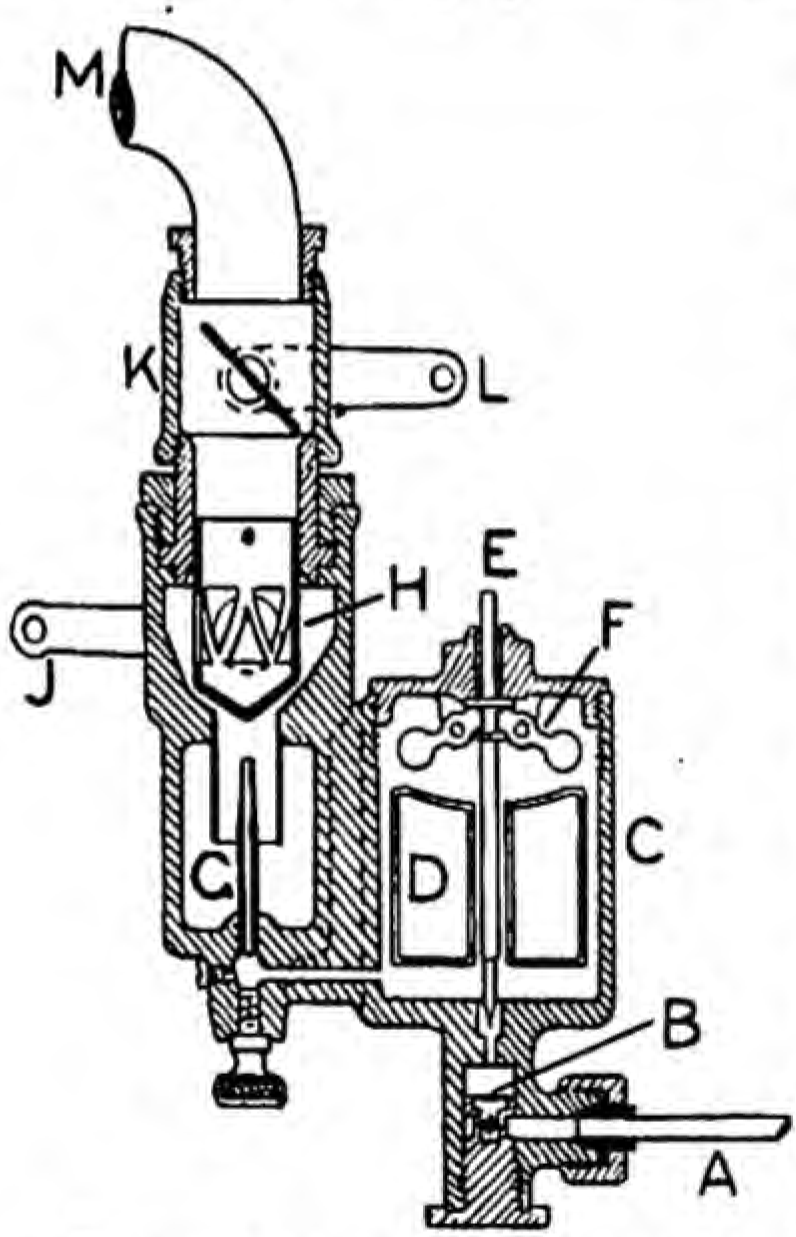


FIG. 321.—Carburettor for a cycle motor.

Carburettor.—Fig. 321 shows in section a carburettor suitable for the motor under consideration. The supply of petrol enters at *A* from a storage tank, and passes through a strainer *B* of fine wire gauze with the object of excluding grit from the fine passages of the carburettor. Having passed the strainer the

petrol enters the feed chamber *C*, which is furnished with an arrangement for controlling the supply. This consists of a pin *E*, pointed to form a valve at its lower end, and having two collars near its top end engaging with two small pivoted levers *F*, weighted so as to hold the pin *E* in the position shown. *D* is a hollow cylindrical float, which will rise when a sufficient quantity of petrol has entered the chamber, and will press the weighted ends of the levers upwards, thus lowering the pin *E* and so closing the inlet orifice at the bottom of the chamber.

The pipe *K* on the engine (Fig. 319) is connected to the pipe *M* (Fig. 321) on the carburettor. During the suction stroke, petrol will flow from the chamber *C* and will be discharged as a fine jet from the vertical tube *G*. This jet is broken up into spray and gasified by the conical baffle plate placed above the tube *G*, and by the rush of air entering through a hole *H* situated on the back of the carburettor. A handle *J* operating a small grating serves to regulate the supply of air. The gasified mixture now passes a butterfly throttle valve *K*, which may be adjusted by means of a handle *L*, and so controls the quantity of mixture to be drawn into the motor cylinder, and finally makes its exit to the cylinder through the pipe *M*. As the carburettor has to be kept warm in order to gasify the spray of petrol efficiently, the connecting pipe to the engine cylinder should be short. Sufficient heat will then be conducted from the cylinder to the carburettor.

Motor car engines.—The drawings given here illustrate the Siddeley 30 horse-power motor car made by the Wolseley Tool and Motor Car Co., Ltd., and are reproduced by courtesy of the makers and of the Editor of "Engineering." The engines are shown in Fig. 322. There are four cylinders arranged vertically in line, side by side The cylinders are each $4\frac{5}{8}''$ bore $\times$ $5''$ stroke and at normal speed of 1000 revolutions per minute give a total of 36 B.H.P. The cylinders are of iron, cast in pairs with the heads and jackets in one piece. It will be noticed that the water jacket covers the inner cylinder heads and extends about half-way down the barrel, thus embracing the whole of the part of the cylinder containing hot gases when the piston is at the bottom of the stroke. The pistons are of the usual elongated type with four spring rings, and are connected direct to the connecting rods by gudgeon pins.

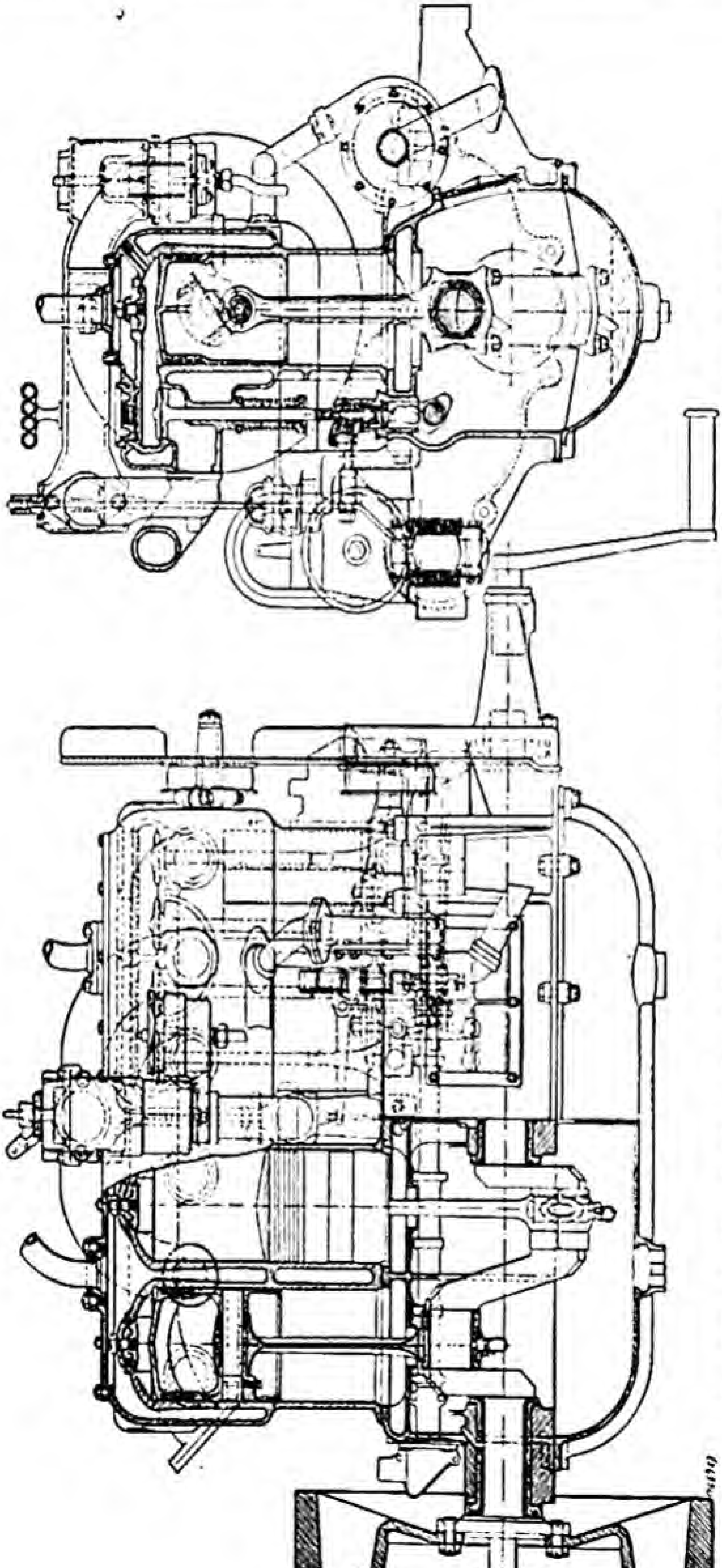


FIG. 322.—Sectional side and end elevations of four cylinder motor for a 30 H.P. Siddeley motor car.

Engine details.—The crank shaft is shown separately in Fig. 323, where it will be noticed that the cranks of the two pairs are set opposite each other. This arrangement gives nearly perfect balance to the whole machine. Reckoning from the forward end of the engine (that end having the handle attached in Fig. 322) the order of firing is: first cylinder, third cylinder, fourth cylinder, second cylinder. As each cylinder has an explosion every two revolutions of the crank shaft, it follows that there will be an explosion and working stroke for each and every half revolution of the crank shaft.

The cylinders are mounted on a base chamber of aluminium, which forms the upper portion of the crank chamber. The lower part of the crank chamber can be removed for inspection. There are in addition two inspection covers on the crank chamber. The

FIG. 323.—Crank shaft for the engines illustrated in Fig. 322.

chamber is in two parts separated by a partition, each part having two cranks working in it. The object is to prevent the oil in the bottom of the chamber (which is used for lubricating the crank pins) from flowing to one end of the chamber when the car is on a gradient and thus cutting off the supply of oil to one or more of the crank pins.

The connecting rods and crank shaft are of forged steel. The crank shaft runs in three main bearings of gun metal bushes lined with white metal. These bearings are lubricated from a reservoir mounted on the dash-board in front of the driver's seat, a separate oil pipe being led to each. The big ends of the connecting rods have short pipes which dip into the oil in the crank chamber at the lowest part of the stroke. The oil thus picked up is driven into the crank pin bearings by centrifugal force.

Valve gear.—The induction and exhaust valves are placed on the same side of the cylinder, and are all on the same side of the engine, thus securing ease in overhauling. These valves are alike

and are interchangeable with each other. They are held down on their seats by helical springs, and are opened at the proper instants by means of cams on a lay shaft, which is contained in the crank chamber. The lay shaft is driven by spur gearing from the crank shaft, and runs at half the speed of the crank shaft. The gearing is situated at the forward end of the engine.

Ignition arrangement.—Ignition is by the high tension magneto system. The magneto-machine is on the same side of the engine as the valves. The firing point is controlled by a lever mounted on the steering wheel of the car. This lever actuates a rocker, which alters the position of the armature in relation to its field, thus timing the spark to suit the speed of the engine. Higher speed requires earlier ignition. An accumulator and high tension coil may also be fitted, and are useful for starting with the engine cold. The carburettor is of special type, and is mounted between the pairs of cylinders, insuring equal feed to each set. A centrifugal governor controls a balanced throttle valve in the inlet pipe, and also regulates the supply of air to the carburettor. The governor can be held out of action by means of a pedal.

Petrol and circulating water supply.—The petrol supply tank has a capacity of 16 gallons; its situation is shown in the plan in Fig. 324. The petrol is fed to the carburettor under pressure, derived from a by-pass on the exhaust pipe while working, and from a hand pump at starting. The supply of circulating water for the jackets is carried in a radiator tank furnished with a number of gilled flattened tubes to provide sufficient cooling surface. The hot water is led from the motor to the tank through a copper pipe, the circulation from tank to motor and back again being maintained by a centrifugal pump (Fig. 322) driven by gearing from the crank shatt. A fan for inducing a draught of cooling air is mounted behind the radiator, and is driven by a flat belt from the crank shaft. The radiator tank forms the front part of the car and carries a bonnet for covering and protecting the engines and gear. The arms of the flywheel (Fig. 322) form vanes, causing a draught of air to circulate round the motors and assist in cooling.

Chassis.—The chassis of the car is shown in Fig. 324. The frame is of pressed nickel steel of tapered channel section without joints. Cross members are riveted to the frame for carrying the

various parts of the mechanism. It will be noticed that all the working parts, excepting those covered by the bonnet, are arranged

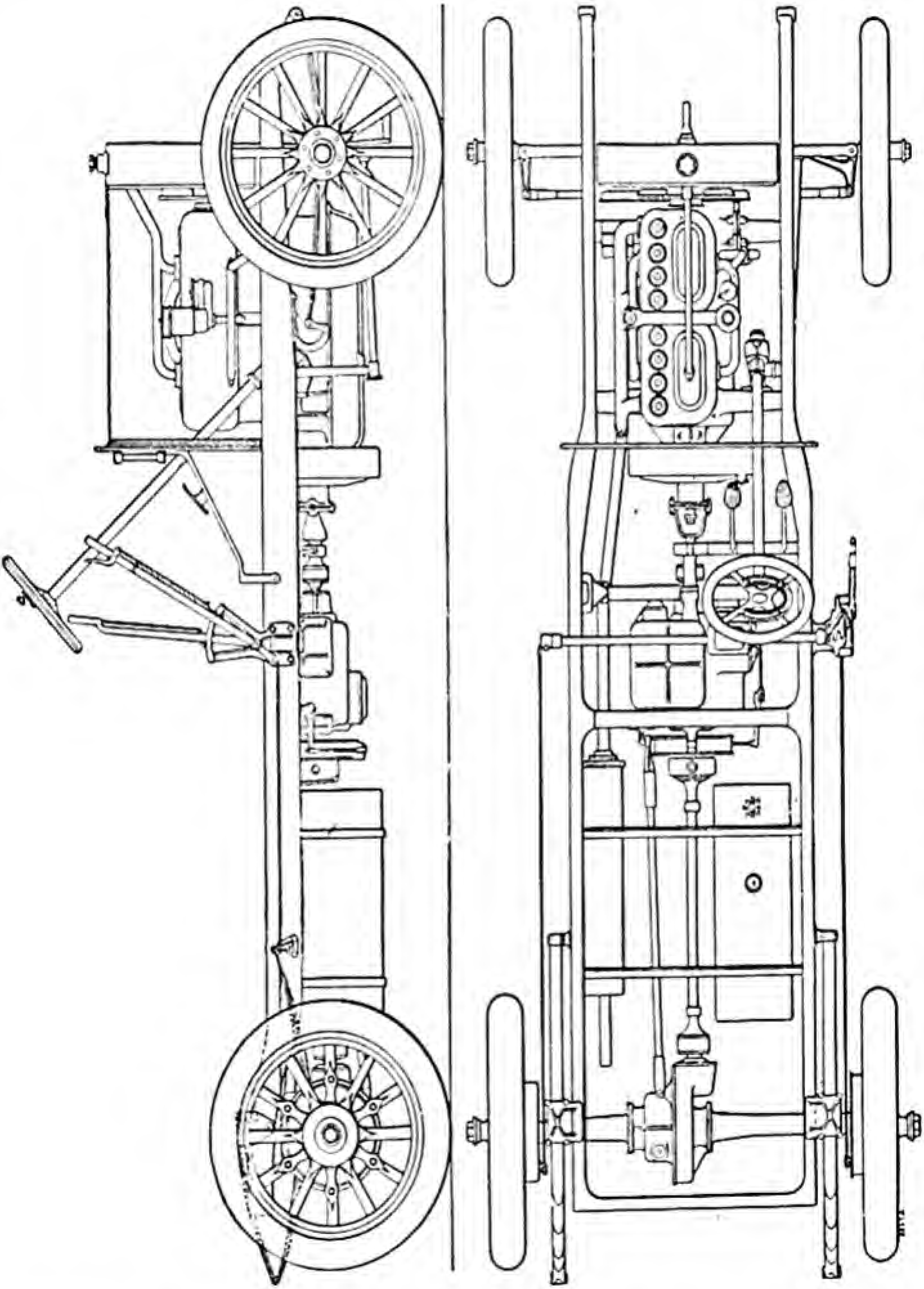


FIG. 324.—Elevation and plan of the chassis for a 30 H.P. Siddeley motor car.

below the frame, thus permitting freedom of design for the car body. The rear wheels are the driving wheels, the front wheels being utilised for steering.

The leading dimensions are :

Normal wheel base, - - - - - - -	10′ 2″
Gauge, - - - - - - - - -	4′ 5″
Distance apart of longitudinal members of frame,	2′ 10″
Dashboard in front of driver to rear end of chassis,	7′ 11″
Dashboard to centre of rear axle, - - - -	7′ 1″
Diameter of wheels, - - - - - - -	34″
Size of pneumatic tyres, - - - - - -	5″

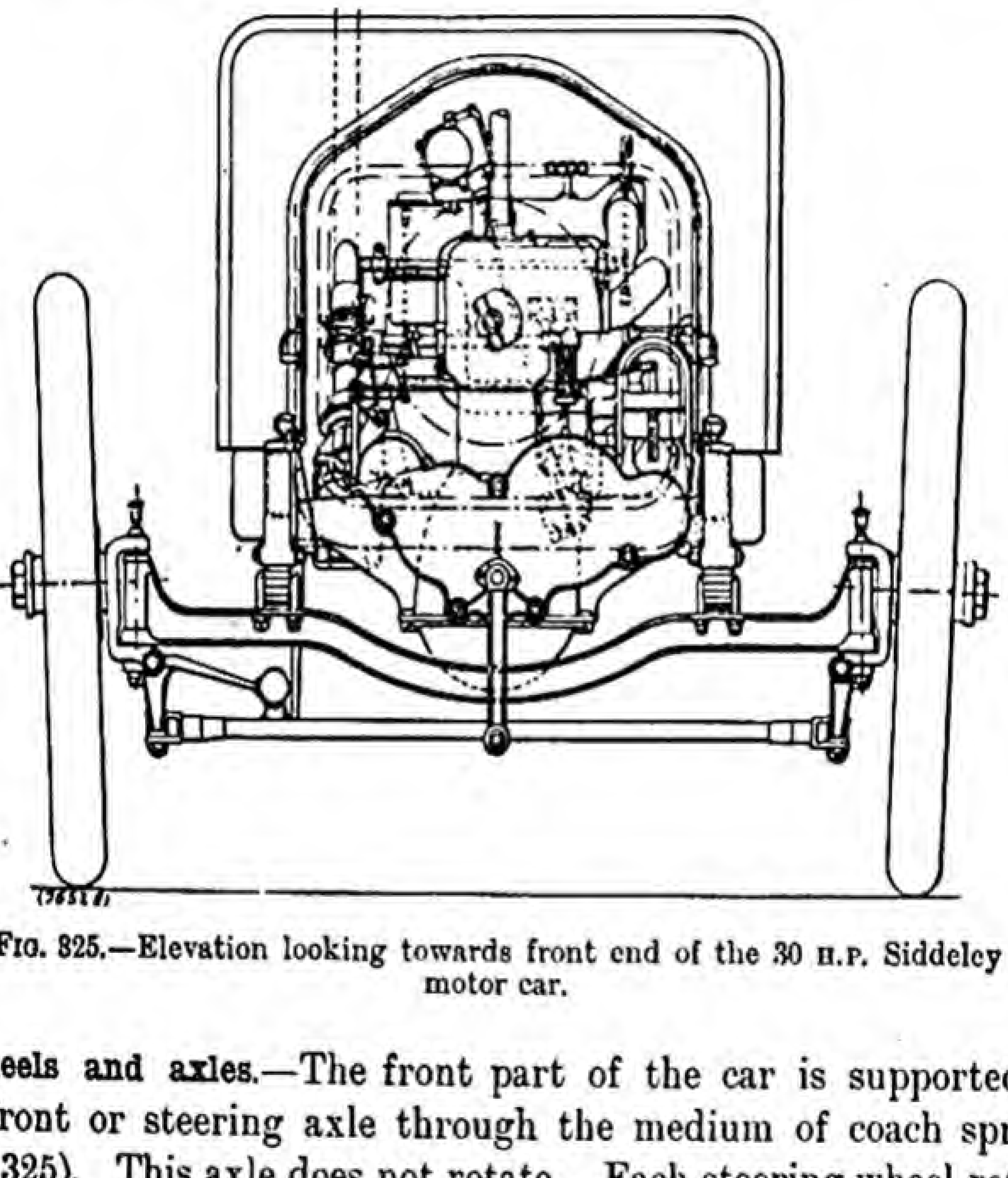


FIG. 325.—Elevation looking towards front end of the 30 H.P. Siddeley motor car.

Wheels and axles.—The front part of the car is supported on the front or steering axle through the medium of coach springs (Fig. 325). This axle does not rotate. Each steering wheel rotates on ball bearings on a swivelling bracket (Fig. 326), which is jawed at its inner end to embrace the end of the steering axle, connection being made by a steel pin. The brackets are connected together by levers and rods so that the wheels may be turned simultaneously in the same direction for steering, being controlled from the driver's seat by a steering wheel. All four wheels are of the artillery pattern. The driving wheels at the rear of the car have their

axle connected direct to three-quarter elliptical springs (Fig. 324), the ends of the springs being connected to the frame of the car. This arrangement is found to give easy riding and flexible drive. The power is transmitted from the motor to the driving wheels by means of a central live axle (*i.e.* an axle capable of rotation) of special hard nickel steel alloy, made by Messrs. Vickers, Sons & Maxim. The makers of the car prefer this arrangement to the alternative one of chain drive.

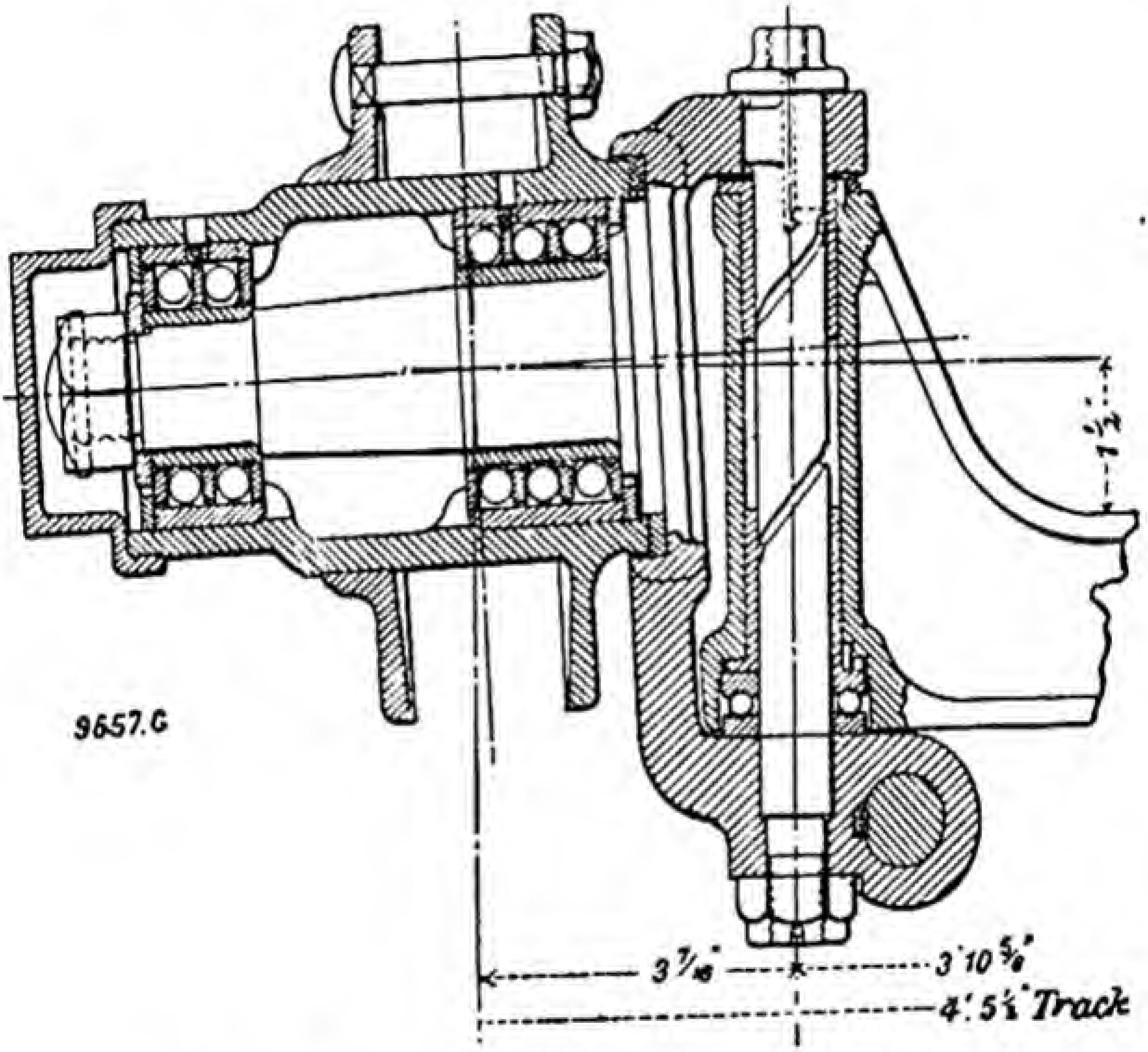


FIG. 326.—Steering swivel of the Siddeley motor car.

The gearing.—The following parts intervene between the motor and the driving wheels: a clutch for disconnecting the motor and allowing it to run free; speed changing gear; bevel gearing and spur differential gearing.

The **clutch** is situated at the rear of the flywheel and is shown in detail in Fig. 327. It is entirely enclosed, is conical in form and is of the metal-to-metal type. The clutch runs in oil to reduce wear. The internal cone is secured to the flywheel, the external cone can slide on the shaft, being controlled by a volute spring which presses at one end against a fixed collar on the shaft and at

the other end against the conical clutch. It will be noticed that this arrangement gives a self-contained thrust, *i.e.* the shaft as a whole does not undergo end thrust. The pressure of the volute spring is adjustable by means of lock nuts screwed to the shaft and bearing against the collar. The clutch is operated from the driver's seat by the connecting levers shown in Figs. 324 and 327.

The clutch is connected flexibly to the first motion shaft in the speed gear box by a short shaft shown in Fig. 327, having universal joints at its ends.

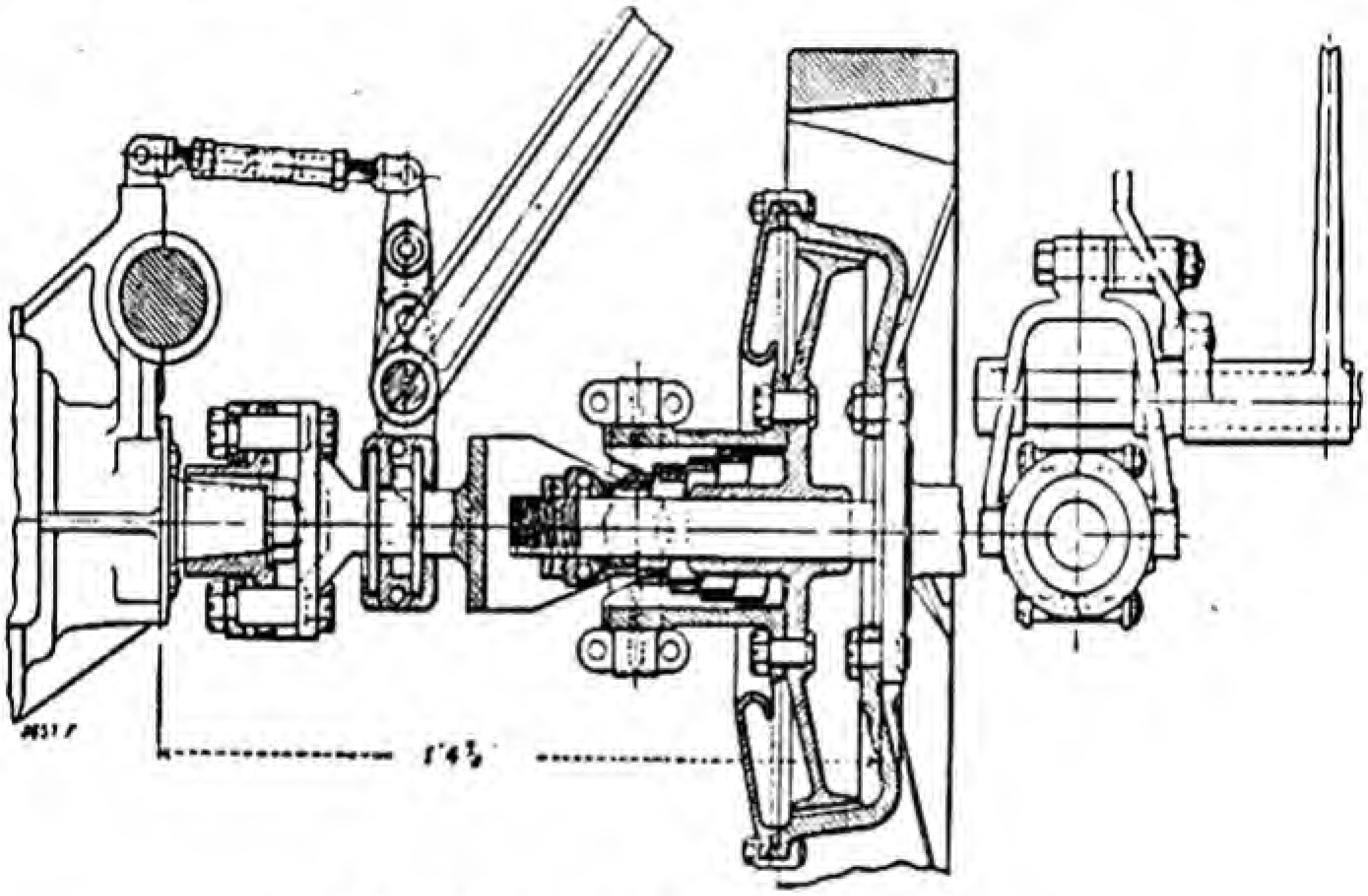


FIG. 327.—Siddeley motor car clutch.

The **gear box** is shown in detail in Fig. 328. The box is mounted on transverse members of the frame. The first motion shaft is made square for a part of its length to receive the speed change wheels which are mounted on it so as to slide, and thus gear as required with a number of wheels mounted on the second motion shaft. There are four forward speeds arranged with normal motor speed of 1000 revolutions per minute, viz. 10, 19, 32 and 40 miles per hour, but these may be varied by altering the speed of the motor. There is also one reverse speed provided for going backwards. Referring to Fig. 328, it will be observed that the flexible shaft from the clutch is connected to a sleeve running in ball bearings, and bored out to form a bearing for the end of the first motion shaft. The sleeve carries a wheel of

24 teeth inside the box gearing with a wheel of 36 teeth on the second motion shaft, which thus will be in rotation constantly if

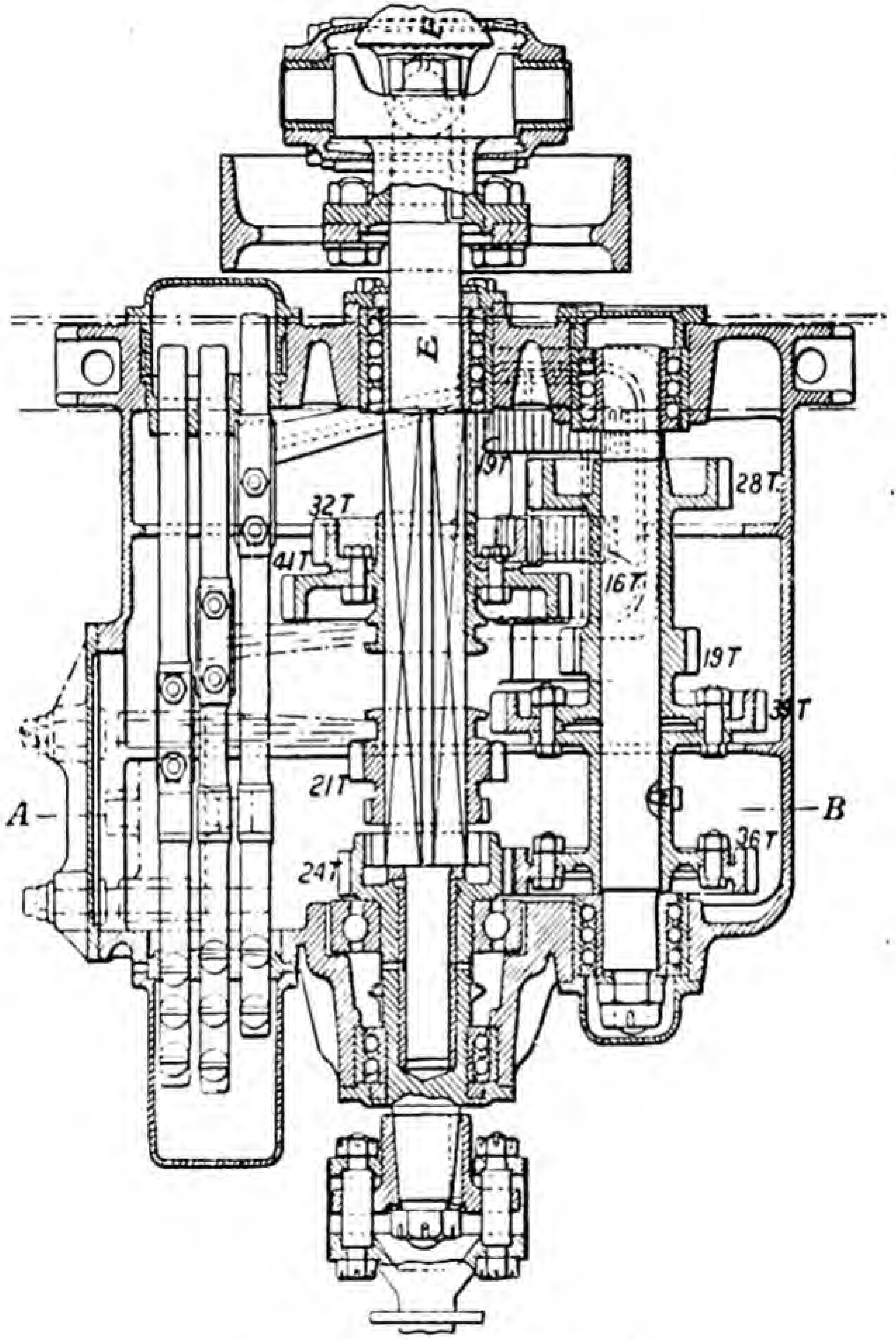


FIG. 328.—Siddeley motor car gear box.

the clutch is in gear. The sliding 21 toothed wheel on the first motion shaft has a toothed clutch, which may enter a corresponding recess with teeth in the 24 toothed wheel at the end of the sleeve. When so geared, the first motion shaft will be driven directly from

the clutch, and will run at the same speed as the motor. This gives a speed of 32 miles per hour to the car. When so geared, the second motion shaft is running idle, none of its wheels being in gear with the sliding wheels on the first motion shaft. The other speeds are obtained by disengaging the internal clutch and sliding the proper wheel on the first motion shaft into gear with the proper wheel on the second motion shaft. Reversal is obtained by the interposition of a pair of idle wheels mounted so as to slide on a third shaft. These have 19 and 16 teeth respectively, and gear thus: the 19 toothed wheel with the 28 toothed wheel on the second motion shaft, and the 16 toothed wheel with the 41 toothed wheel on the first motion shaft. The gearing shafts run on multiple ball bearings at each end. The gearing wheels are made of Vickers steel alloy, having a strength of 80 tons per square inch after case hardening.

Speed gear arrangements.—The gear wheels are slid into position by levers operated by striking rods shown in Fig. 328. The striking rods are controlled by a bent lever, connected to a hand lever at the driver's seat, the latter lever working in a notched gate. The striking rods are on the interlocking system, resembling railway signals. When one striker is in gear it holds a plate locking the other strikers. A spring-controlled plunger drops into one or other of the notches in this plate and holds the plate until disengaged by the striker being thrown over, when the plate may slide into the next position.

The gear box is connected to the **bevel gear** by means of a shaft with universal joints at each end. The drive is taken by the bevel gear (Fig. 329) direct to both portions of the driving axle, these being connected by **spur differential gearing**. The function of the latter gear is to allow of one portion of the driving axle with its wheel to run at a slightly different speed to the other portion when rounding a curve or bend on the road.

Driving wheels and axle.—The driving axle is placed inside a tubular casing on which the weight of the car is supported through the three-quarter elliptical springs. The hubs of the driving wheels are mounted on a square end of the axle and so have the drive communicated to them. The hubs run on multiple ball bearings, which take shocks, due to the inequalities of the road. The arrangement also permits of bearings of large diameter and

length being used. Practically all of the bearings outside of the motor are ball bearings, thus securing a high mechanical efficiency.

Brakes.—A torque bar is carried from the rear axle to a bracket suspended to the main cross member of the frame. This bar is flexible in all directions excepting in that of rotation and takes the

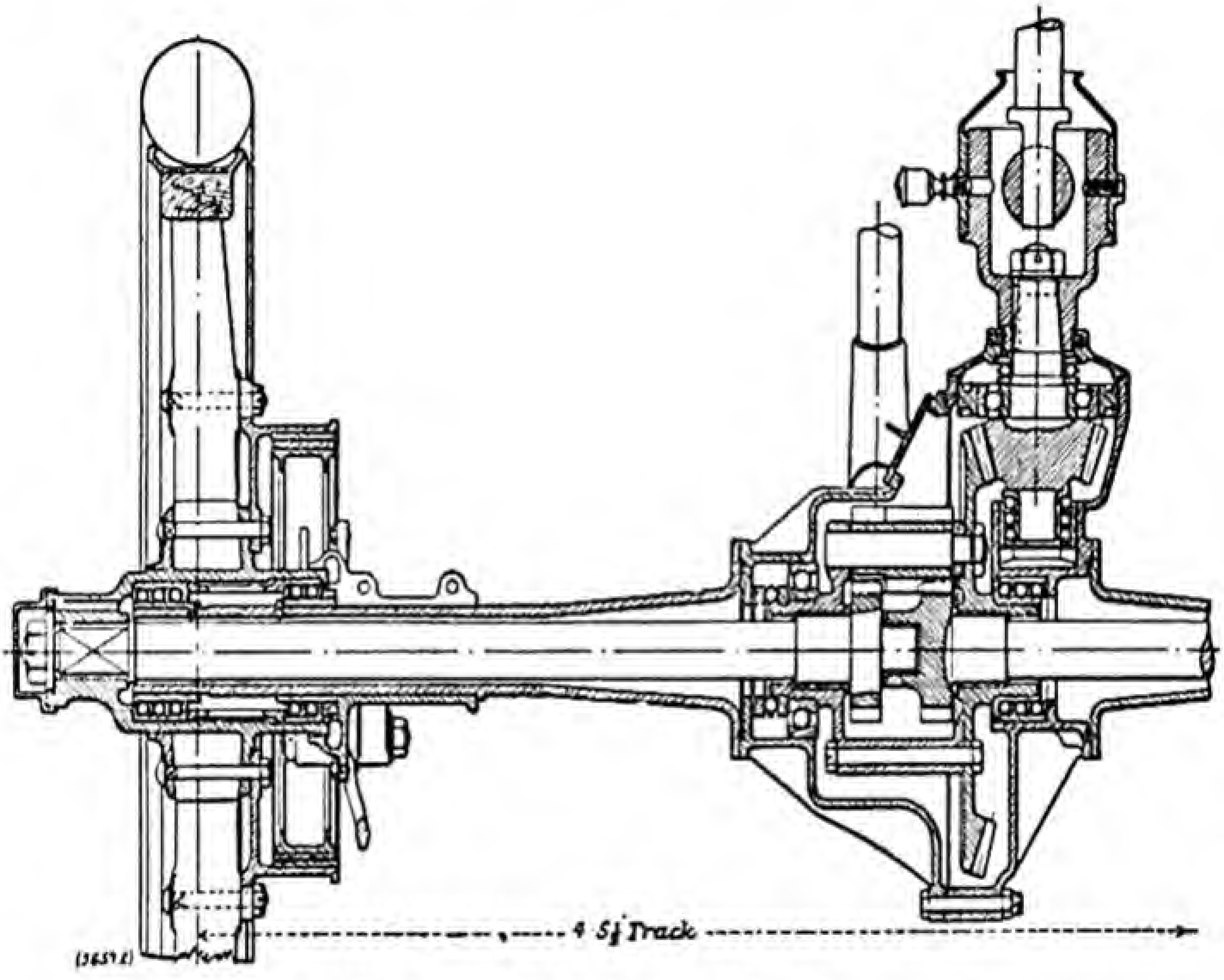


FIG. 329.—Differential gear and driving axle of the Siddeley motor car.

whole of the reaction, both for driving effort and for brake application. A foot brake is fitted to the driving shaft at the rear end of the gear box. This is of the band type, and means of easy adjustment are provided. There are also brake drums on each rear wheel. These are entirely enclosed, the brakes being of the internal expanding type, and are operated by a hand lever to which they are connected by wire cables. The latter brakes are so arranged that they are brought into operation simultaneously and equally.

EXERCISES ON CHAPTER XX.

1. Describe generally the action in any oil or petrol engine you know.

2. Explain with sketches what must be done to the liquid fuel before being mixed with air.

3. Sketch in section an oil engine cylinder showing the valves. Name and explain the use of each part.

4. Describe and give sketches of an oil engine governor.

5. Explain with sketches how the charge may be ignited in an oil or petrol engine.

6. Why is a single cylinder engine seldom found in a modern motor car?

7. Explain any system of lubrication suitable for a motor car engine.

8. Sketch, with as little detail as possible, the plan of the chassis of a motor car. Name each part, and explain how the motion is transmitted from the engine crank shaft to the driving wheel.

9. Explain, with sketches, how a motor car is steered.

10. Describe, with sketches, any mechanism for varying the speed of a motor car.

11. What is the function of the differential gearing of a motor car? Give sketches and describe any such arrangement.

CHAPTER XXI.

GAS AND OIL ENGINE TESTS.

Tests on internal combustion engines.—Many instructive experiments may be carried out by students should a gas or oil engine be available. A few of these are here described.

Brake horse-power.—To obtain the Brake Horse-Power of a small gas or oil engine such as is generally installed in engineering laboratories, any of the ordinary forms of rope brake may be used. When working at low powers, it will sometimes be found convenient to have the rope lapping one half only of the circumference of the flywheel. The revolutions should be counted throughout the whole test by a revolution counter driven from the crank shaft; frequent checking by counting the revolutions during one minute at different parts of the run is desirable in order to make sure that the speed of the engine is not varying considerably. The latter readings should be compared with the average number found by dividing the total revolutions for the run shown on the counter by the duration of the test in minutes. The Brake Horse-Power will be calculated as described in Chap. VII., and the result, if the test is carefully carried out, will be a very close approximation to the actual horse-power which the engine is capable of delivering.

Indicated horse-power.—The Indicated Horse-Power of an internal combustion engine can only be roughly estimated. This arises from the different shapes of diagram produced even when the engine is working as steadily as can be obtained. The difficulty is increased greatly should the engine be governed by reducing (not cutting off altogether) the supply of fuel. In such cases it is

almost impossible, with an ordinary indicator, even roughly to estimate the average diagram.

To secure the best results, the indicator employed should be of light construction in its working parts, and a strong spring should be used. Should this not be attended to, the impulse given to the indicator piston when explosion occurs will set up vibrations, which will be communicated to the indicator pencil and give a series of waves instead of the true expansion curve.

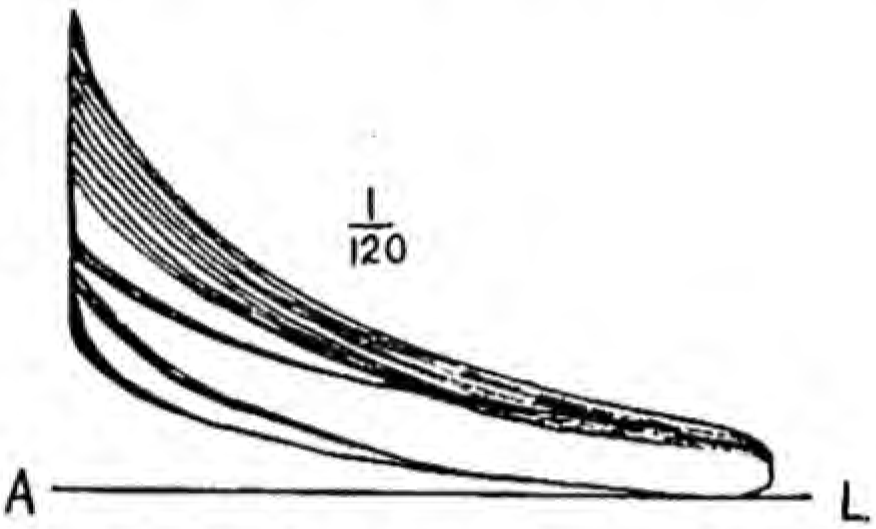


FIG. 330.—26 cycle indicator diagram from an oil engine governed by adjusting strength of mixture.

Fig. 330 shows a card taken from an oil engine governing by regulating the supply of oil so as to vary the strength of the explosive mixture. The engine was running at 208 revolutions per minute, and the pencil was held against the paper during ¼ minute. There are thus 26 cycles on the diagram. The difficulty of obtaining the indicated horse-power may be imagined. To throw light on the manner in which the explosions follow one another, the diagram shown in Fig. 331 was taken as rapidly as possible after that first taken. The cord driving the indicator drum was disconnected from the engine and drawn very slowly by hand during ¼ minute while the pencil was in contact with the paper. The whole of the 26 cycles was thus compressed into the length of the diagram, each cycle being represented by a very narrow strip. From the heights of these, the maximum pressures may be measured, and so some notion as to the strength of the mixture arrived at. The diagram corresponding

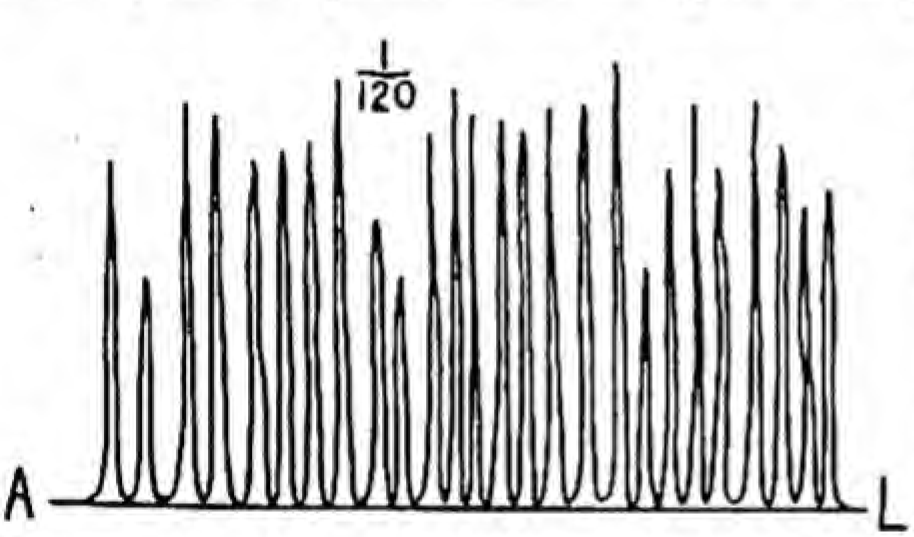


FIG. 331.—26 cycle explosion diagram corresponding to Fig. 330.

to any particular cycle in Fig. 331 may be found by measuring its height and taking the diagram showing the same height in Fig. 330.

To arrive approximately at the average indicated horse-power developed during a test, a number of diagrams should be taken, each showing say ten cycles. Using a piece of tracing paper, an average diagram may be guessed. Suppose this to be done, and that the diagram resembles that shown in Fig. 332. During the compression stroke, work is done on the mixture as represented by the area from PV up to the compression curve; during the

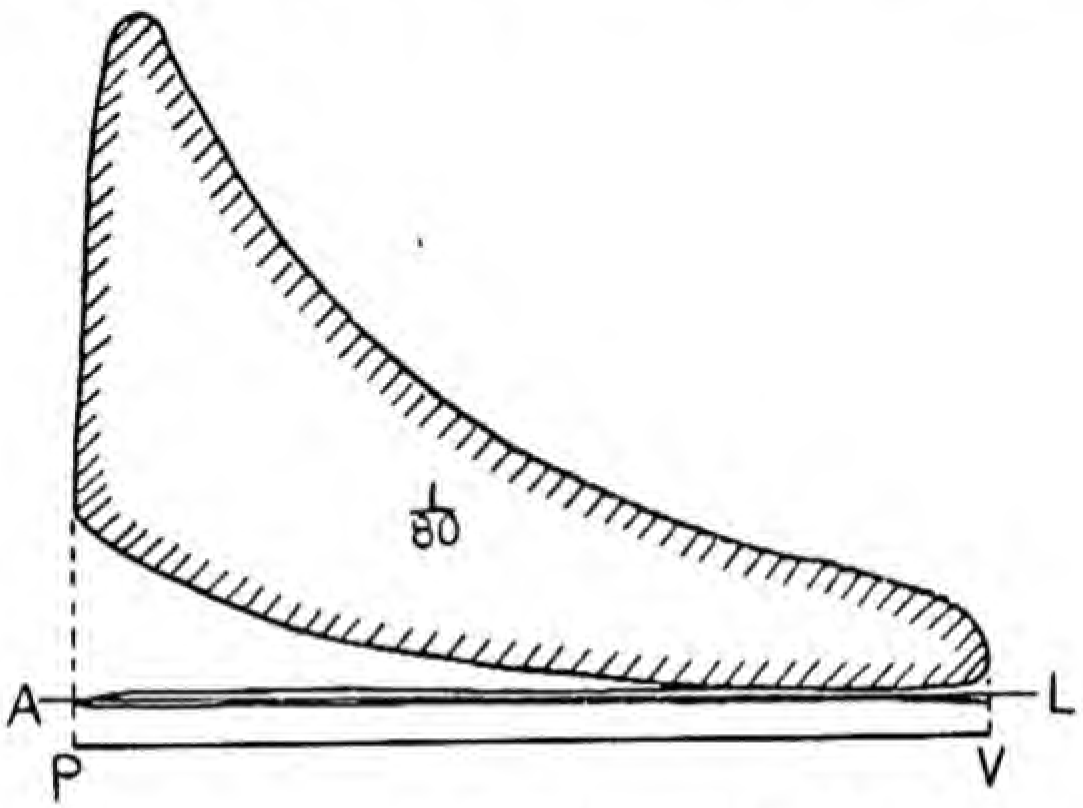


FIG. 332.—An averaged indicator diagram from a gas engine.

expansion stroke, work is done on the piston as represented by the whole area from PV up to the expansion curve. It therefore follows that the net work done on the piston will be represented by the closed shaded area. The average pressure, p_m, may be found from this area by any of the usual rules described in Chap. VII.

Having thus found the average mean pressure for the run, we must also know how many explosions occurred per minute. In the case of an engine governing by **hit or miss**, the explosions per minute must be counted systematically during the test. This may be done either by listening for the sound of the explosion, or by watching the stem of the intake valve. In the case of an engine governing by varying the fuel supply, the number of explosions

per minute may be taken as equal to half the number of revolutions per minute for the Beau-de-Rochas cycle.

Let p_m = average mean pressure, lbs. per sq. inch,

A = area of the piston, sq. inches,

L = length of stroke, feet,

E = number of explosions per minute;

then,

$$\text{I.H.P.} = \frac{p_m \times A \times L \times E}{33,000}.$$

The most useful practical information given by the indicator diagram is that concerning the manner in which the cycle is being carried out by the various valves. A glance at the diagram by an experienced eye at once tells whether all is as it should be or whether one or more of the valves is out of time, or is leaking.

Measurement of heat supplied to a gas engine.—In the case of a gas engine, the supply of gaseous fuel is usually measured by means of a dry gas-meter. The meter is best read as follows: During the run, the observer should note the time (hour, minute, and second) of each successive passage of the pointer on the meter over the zero mark. Suppose one revolution of the pointer to indicate that 10 cubic feet of gas have passed, and that the observer has noted that the time of revolution of the pointer is t minutes. Then,

$$\text{Gas consumed per minute} = \frac{10}{t} \text{ cubic feet.}$$

The gas consumed per minute throughout the run will thus be known. This method is preferred because the mechanism of the meter will be always in the same position and under the same conditions each time a reading is taken, and errors are thus eliminated. The gas-meter should be proved occasionally at the official testing house.

Reduction to standard temperature and pressure.—In order that tests on gas engines may be comparable, the meter reading should be reduced to standard temperature and pressure. To enable this to be done, the following observations must be made:

(1) Temperature of gas supply, t° C.

(2) Pressure of gas supply, h millimetres of water.

(3) Barometric pressure during test, H millimetres of mercury,

The temperature of the gas may be measured by having a tee-piece on the gas pipe between the meter and the antifluctuator, and inserting a thermometer pushed through a cork fitted to the branch on the tee-piece. The pressure above that of the atmosphere may be measured at the same place by means of a glass U-gauge containing water, one limb being connected to the gas supply pipe and the other left open to the atmosphere. The pressure will be measured in millimetres of water by means of a scale placed between the branches of the U-gauge, the difference in the heights of the water columns being noted. This may be converted to millimetres of mercury by division by 13·6, the specific gravity of mercury.

Taking standard temperature and pressure to mean 0° C. and 760 mm. mercury column, the required reduction would be thus calculated:

Absolute pressure of gas $=\left(H+\frac{h}{13 \cdot 6}\right)=h_1$ millimetres of mercury.

Absolute temperature of gas $=(273 \cdot 7+t)$ Centigrade.

Let $V_1=$ volume in cubic feet per minute by meter,

$V=$ volume of this at standard temperature and pressure.

Applying the general law

$$\frac{P_1 V_1}{\tau_1}=\frac{P_2 V_2}{\tau_2} \text{ (p. 54),}$$

we obtain

$$\frac{h_1 V_1}{273 \cdot 7+t}=\frac{760\ V}{273 \cdot 7}$$

or,

$$V=\left\{\frac{273 \cdot 7 \times h_1}{760(273 \cdot 7+t)}\right\} V_1 \text{ cubic feet.}$$

The heat supplied in this calculated volume of gas can be determined, provided the heating value per cubic foot has been first determined or is otherwise known.

Let $Q=$ heating value of the gas, in B.T.U. per cubic foot,

then Heat supplied per minute $=(Q \times V)$ B.T.U.

Measurement of heat supplied to an oil engine.—This can be easily effected by disconnecting the ordinary supply tank and allowing the engine to draw its supply of fuel from a can of sufficient capacity. A mark should be made so that the can may

be filled to the same level when required. The best and simplest way is to lay a piece of wood across the top of the can; a sharp-pointed stiff wire hook projecting downwards from the piece of wood will indicate the instant when the standard oil level is reached by its point just breaking the surface of the oil. At the instant the engine test is begun, the level of the oil in the can should be brought to standard level, and at the moment that the test is over the level should be again raised to standard by pouring in a measured quantity of oil. The weight of the oil poured in thus to restore the original level is equal to that used by the engine during the test.

Let W = weight of oil used per minute, lbs.,

Q = heating value of oil, in B.T.U. per lb.,

then Heat supplied per minute $= (W \times Q)$ B.T.U.

Heat absorbed by the jacket water.—In ordinary internal combustion engines the circulation of the jacket water is maintained, as already described, by means of natural gravitational currents of water or by forced circulation from a pump. In measuring the heat absorbed by the jacket water it is necessary to measure the inlet and outlet temperatures of the water and also the quantity of water passed in a given time. To enable these observations to be taken, it is best to arrange the engine so as to draw a constant supply of water to the jacket from a tank and to discharge it into another tank, where it can be measured or weighed. Thermometers pushed through corks and inserted in tee-pieces on the inlet and outlet water pipes close to the engine serve to measure the temperatures.

Let W = weight of water passed in a time t minutes,

t_1 = inlet temperature, Fah.,

t_2 = outlet temperature, Fah.

$$\text{Total heat absorbed} = W(t_2 - t_1), \text{ B.T.U.}$$

$$\text{Heat absorbed per minute} = \frac{W(t_2 - t_1)}{t}, \text{ B.T.U.}$$

Heat balance.—Account can now be made of the heat supplied and how it disappears. The quantities known are, I.H.P., B.H.P., the heat supplied per minute to the engine $= H$, and the heat absorbed by the jacket water per minute.

Energy given to the piston per minute = I.H.P. × 33,000 ft.-lbs.

Taking $J = 774$, this will give the energy transferred to the piston per minute

$$=\left(\frac{\text{I.H.P.} \times 33{,}000}{774}\right), \text{ B.T.U.}$$

Useful energy produced by the engine per minute

$$= \text{B.H.P.} \times 33{,}000 \text{ ft.-lbs.}$$

$$=\left(\frac{\text{B.H.P.} \times 33{,}000}{774}\right), \text{ B.T.U.}$$

Energy wasted per minute in overcoming engine friction

$$=(\text{I.H.P.} - \text{B.H.P.})\frac{33{,}000}{774}, \text{ B.T.U.}$$

Let energy absorbed per minute in heating jacket water

$$= E_j, \text{ B.T.U.}$$

The remainder of the energy is wasted by incomplete combustion, by heat carried away in the exhaust gases, and in other ways. The proportion due to each is not easily determined, and we may class them all under one head and say,

Let energy wasted in exhaust, etc. $= E_x$, B.T.U. per min.

A balance sheet can now be made out, and for this purpose it is convenient to express each of the above quantities as a percentage of H = the heat supplied per minute. Reckoning this energy supplied as 100, the others will become:

$$\text{Energy given to the piston} = \frac{\text{I.H.P.} \times 33{,}000}{774 \times H} \times 100.$$

$$\text{Useful energy obtained} = \frac{\text{B.H.P.} \times 33{,}000}{774 \times H} \times 100.$$

Energy wasted in engine friction

$$= \frac{(\text{I.H.P.} - \text{B.H.P.})33{,}000}{774 \times H} \times 100.$$

Energy wasted in heating the jacket water

$$= \frac{E_j}{H} \times 100.$$

Energy wasted in heating exhaust gases, etc.

$$=\frac{E_x}{H}\times 100.$$

HEAT ACCOUNT.

By energy supplied, -	100	To useful energy obtained,	17·2
		,, energy wasted in friction, - - - -	4·3
		,, energy wasted in heating water, - -	33·2
		,, energy wasted in heating exhaust gases, etc. (by difference), -	45·3
	100		100

The numbers supplied in the above account for the sake of illustration were obtained in a test of a gas engine.

Gas engine tests.—In testing a gas engine for B.H.P., I.H.P. and fuel consumption, the following observations must be made:

1. Indicator diagrams to be taken every 10 minutes. About 10 cycles should appear on each.
2. Revolutions per minute of the crank shaft.
3. Explosions per minute.
4. Brake spring balance and dead load readings taken every 10 minutes and frequent intermediate check readings made.
5. Gas passed by meter, read by noting the times to the nearest second, at which the index passes zero throughout the test.
6. Temperature of the gas, every 10 minutes.
7. Pressure of the gas, every 10 minutes.
8. Reading of the barometer. This need only be taken once for each test of one hour's duration.
9. Time taken to pass a measured quantity of water through the cylinder jacket.
10. Inlet and outlet temperatures of the circulating water.
11. Temperature of the engine room.

Measurements should be taken from the engine of the following particulars :

PRINCIPAL DIMENSIONS AND CONSTANTS.

1. Diameter of the piston, d inches, - - - ....
2. Stroke of the piston, L feet, - - - - ....
3. Diameter of the brake wheel to the centre of the rope, D feet, - - - - - - ....

VALVE SETTINGS.

Crank angle at which :	Gas inlet valve.	Air inlet valve.	Exhaust valve.	Ignition valve.
Valve begins to open, -				
Valve full open, - -				
Valve begins to close, -				
Valve closed, - - -				

The I.H.P. and B.H.P. constants should be worked out in order to facilitate subsequent calculation.

Let p_m=average pressure from the diagrams,

E=explosions per minute,

then
$$\text{I.H.P.}=\frac{p_m\times\frac{\pi d^2}{4}\times L\times E}{33{,}000}$$
$$=\left(\frac{\pi d^2\times L}{4\times 33{,}000}\right)\times p_m\times E$$
$$=c\times p_m\times E, \quad\ldots\ldots\ldots\ldots(1)$$
in which c, the bracketed quantity, is the I.H.P. constant for the engine. The value of c should be calculated from
$$c=\frac{\pi d^2\times L}{4\times 33{,}000}$$
and inserted in equation (1).

Let N=revolutions per minute,

$(W-S)$=net brake load, lbs.,

then
$$\text{B.H.P.}=\frac{(W-S)\times N\times\pi D}{33{,}000}$$
$$=\left(\frac{\pi D}{33{,}000}\right)\times(W-S)N$$
$$=K(W-S)N \quad\ldots\ldots\ldots\ldots(2)$$

Calculate the value of K from

$$K=\frac{\pi D}{33{,}000}.$$

This result will give the B.H.P. constant; insert its value in equation (2).

Logs of test.—The readings taken during a test are best entered in a tabular form. To illustrate the method the following example of a gas engine trial is included.

Example of gas engine test.—The engine tested is a $6\frac{1}{2}$ B.H.P. Crossley engine, supplied by the makers in 1898 to the author's laboratory. Its principal work has been the driving of machinery, and, when required for testing, the belt connecting it to the main shaft is removed. It must be understood that the engine was in no way specially adjusted before any tests, and therefore the results obtained are such as would be produced at any time by the engine while performing ordinary driving work. The engine has been fully described in Chap. XIX.

PRINCIPAL DIMENSIONS AND CONSTANTS.

1. Diameter of the piston, d inches, - - - - 6·69
2. Stroke of the piston, L feet, - - - - - 1·187
3. Diameter of the brake wheel to the centre of the rope, D feet, - - - - - - - 4·719

VALVE SETTINGS.

Crank angle at which:	Gas inlet valve.	Air inlet valve.	Exhaust valve.	Ignition valve.
Valve begins to open, -	358°	320°	123°	299°
Valve full open, - -	70°	70°	236°	$2\frac{1}{2}$°
Valve begins to close, -	99°	99°	260°	325°
Valve closed, - - -	180°	220°	$12\frac{1}{2}$°	104°

Constants.—I.H.P. constant $=c=\dfrac{\pi d^2\times L}{4\times 33{,}000}$

$$=\frac{35\cdot14\times1\cdot187}{33{,}000}$$

$$=0\cdot00126;$$

$$\therefore\ \text{I.H.P.}=0\cdot00126\times p_m\times E.$$

$$\text{B.H.P. constant} = K = \frac{\pi D}{33{,}000}$$
$$= \frac{14{\cdot}83}{33{,}000} = 0{\cdot}000448\,;$$
$$\therefore\ \text{B.H.P.} = 0{\cdot}000448 \times (W - S) \times N.$$

Gas Engine Test.

No. of test: 1.

Description of engine: $6\frac{1}{2}$ B.H.P. Crossley gas engine.

Where situated: Engineering Laboratory, West Ham Technical Institute.

Date of test: Dec. 16th, 1898.

Engine Log.

Time.	Revs. per minute.	Brake.		No. of indicator diagram.	Explosions per minute.
		Spring balance.	Dead load.		
p.m.		S lbs.	W lbs.		
8.0	212	$10\frac{3}{4}$	81	1	70
8.10	213	$10\frac{1}{2}$	81	2	70
8.20	216	$10\frac{1}{2}$	81	3	70
8.30	215	$12\frac{1}{2}$	81	4	70
8.40	215	$11\frac{1}{2}$	81	5	70
8.50	215	$11\frac{3}{4}$	81	6	70
9.0	214	$11\frac{3}{4}$	81	7	70

Gas Log.

Time at which index passes zero.	Quantity of gas measured in interval.	Temperature of gas.	Pressure of gas.	Barometer.
m. s.	cubic feet.	F.	mm. of water.	mm. of mercury.
0 0				
3 25	10	69°	118	765
6 50	10	70°	114	,,
10 30	10	70·5°	118	,,
14 22	10	69·5°	114	,,
17 50	10	69°	114	,,
21 16	10	69°	114	,,

Circulating Water Log.

Time.	Water passed through jacket.	Inlet temperature.	Outlet temperature.
h. m. s.	lbs.	F.	F.
8 0 5	0	48°	96·5°
8 6 45	100	48°	97·5°
8 14 0	200	48°	97°
8 21 35	300	48°	98°
8 29 0	400	48°	98°
8 36 25	500	48°	98°
8 44 5	600	48°	98°

Results of Gas Engine Test.

No. of test,	1
Date of test,	Dec. 16th, 1898.
Duration of test,	1 hour
Average revs. per minute,	215
Brake, average dead load, W lbs.,	81
average spring balance pull, S lbs.,	$11\frac{1}{4}$
net average brake load, $(W-S)$ lbs,	68·75
Brake horse-power,	6·63
Average mean pressure from indicator diagrams, p_m lbs. per sq. inch,	87·1
Average explosions per minute,	70
Indicated horse-power,	7·7
Volume of gas as measured by meter, cubic feet per hour,	169
Average pressure of gas, mm. of water,	115
Barometric pressure, mm. of mercury,	765
Absolute pressure of gas, mm. of mercury,	773·5
Average temperature of gas, F.,	69·5°
Volume of gas used in cubic feet per hour at 0° C. and 760 mm. of mercury pressure,	160
Volume of gas in cubic feet at 0° C. and 760 mm. of mercury pressure per B.H.P. per hour,	24·2

Heat disposal.

Heating value of gas in B.T.U. per cubic foot at 0° C. and 760 mm. mercury pressure,	620
Heat supplied to engine per minute, B.T.U.,	1650
Heat equivalent to useful work done per minute, B.T.U.,	282
Work done per minute in driving the engine mechanism, ft.-lbs.,	35,400
Heat equivalent to the work done per minute in driving the engine, B.T.U.,	45·6
Weight of circulating water passed per min., lbs.,	13·64
Average rise in temperature of circulating water, degrees Fah.,	49·5
Heat carried away in jacket water per minute, B.T.U.,	675

HEAT ACCOUNT.

By energy supplied,	100	To useful energy obtained,	17·1
		,, energy wasted in friction,	2·76
		,, energy wasted in heating water,	40·9
		,, energy wasted in exhaust gases, etc. (by difference),	39·24
	100		100·00

Temperature at points in the gas engine cycle.—It may be assumed, in making this calculation, that the gaseous mixture of gas and air present in the cylinder at the end of the suction stroke follows the law

$$\frac{pv}{\tau} = c. \qquad \text{(p. 54.)}$$

It thus becomes possible to calculate approximately the temperatures at other points of the compression and expansion strokes. To enable the calculation to be made, the pressure, volume and temperature of the inclosed mixture must be known for one point,

and also the pressure and volume for the point at which the temperature is required.

EXAMPLE. In a gas engine cylinder where $v=2\cdot2$ and $p=14\cdot72$, it was known that the temperature was 130° C. What is the temperature when $p=122$ and $v=1\cdot2$? 1906.

The initial temperature $=130+273\cdot7$

$$=403\cdot7^\circ \text{ C. absolute.}$$

$$\frac{p_1v_1}{\tau_1}=\frac{p_2v_2}{\tau_2}$$

$$\therefore\ \tau_2=\frac{p_2v_2\tau_1}{p_1v_1}$$

$$=\frac{122\times1\cdot2\times403\cdot7}{14\cdot72\times2\cdot2}$$

$$=1826^\circ \text{ C. absolute,}$$

$$t=1826-273\cdot7=\underline{1552}^\circ \text{ C.}$$

EXERCISES ON CHAPTER XXI.

1. Explain fully the reasons why the B.H.P. of an internal combustion engine can be stated more accurately than the I.H.P.

2. State the measurements which must be taken in testing a gas engine or an oil engine for fuel consumption per B.H.P. per hour. What precautions are necessary?

3. In a gas engine test the gas supplied, as measured by meter, was 87 cubic feet at $1\frac{1}{2}''$ water pressure and 15° C., this quantity being supplied in 52 minutes. Calculate what quantity of gas per hour at standard temperature and pressure entered the engine. The barometer during the test showed $29\frac{1}{2}$ inches of mercury pressure.

4. Describe how the heat carried away by the jacket water may be measured. Sketch the arrangement.

5. In an oil engine trial, the circulating water entered at 50° F. and was discharged at 95° F. Quantity used, 450 lbs. in 32 minutes. Calculate what heat per minute is being carried away by the water. To how much mechanical work is this equivalent?

6. Describe, without too much detail, the working of any gas or oil engine with which you are acquainted.

If 20 lbs. of oil (calorific value 21,000 Fahrenheit units) are used per hour, the brake horse-power being 18, what is the efficiency? 1898.

7. An engine gives 10 indicated horse-power and 7·6 brake horse-power for a consumption of 230 cubic feet of coal gas per hour. The calorific power of the gas is 530,000 foot-pounds per cubic foot. What is the efficiency? 1897.

8. Sketch a gas engine indicator diagram. How is it used in finding the indicated horse-power? State clearly what information is necessary. Why must we know the number of explosions per minute rather than the number of revolutions? 1900.

9. Draw the compression, ignition, and expansion part of a gas engine diagram. If the volumes and pressures at four points on the diagram, to any scales whatsoever, are represented by

Points, - -	*A*	*B*	*C*	*D*
Volumes, -	5	1·5	2	4
Pressures, -	1	4	10	3

And if at the point *A* we know that the temperature is 127° C., what are the temperatures at the other points? Tabulate the results. 1901.

CHAPTER XXII.

HISTORICAL NOTES.

Hero's engine.—The earliest recorded steam engine was described by Hero of Alexandria more than 2000 years ago. This engine was a form of steam turbine, working by the reaction of steam jets discharged from orifices formed in a vessel free to rotate. It is not until the 17th century that we find inventors seriously attempting to produce commercial engines, and to Savery is due the credit of the first successful steam engine.

Savery's engine.—Savery's engine, patented in 1698, was devised for pumping water from mines. Its action will be understood by reference to Fig. 333. A and B are two vessels connected at their tops to a steam pipe leading from a boiler; S_1 and S_2 are steam valves which may be opened and closed by hand. The vessels are connected at the bottom to two branches of a water pipe, the lower part of which is carried down the mine to the water to be drained, the upper part leads to the surface level where the water is to be discharged. V_1 and V_2 are non-return suction valves; V_3 and V_4 are non-return discharge valves. Steam is supplied from a boiler C, a second boiler D being used to supply C with feed water.

The vessels A and B were used alternately in the following manner. S_1 being opened. steam entered A and drove out any air. The steam valve was then closed and a jet of water was allowed to play on the outside of the vessel A, thus causing the steam to condense and forming a partial vacuum. Water ascended the suction pipe, passing the valve V_1 and entered A. S_1 being now opened, the steam by its pressure on the water in A caused it to flow through the valve V_3, and so up the discharge pipe. In actual

working, *A* was filling while *B* was discharging, and *vice versâ*, a fairly constant discharge being thus obtained. The height to which water could be forced depended, of course, on the steam pressure available, which, owing to imperfections in boiler construction at that time, was such that a maximum lift of 80 feet

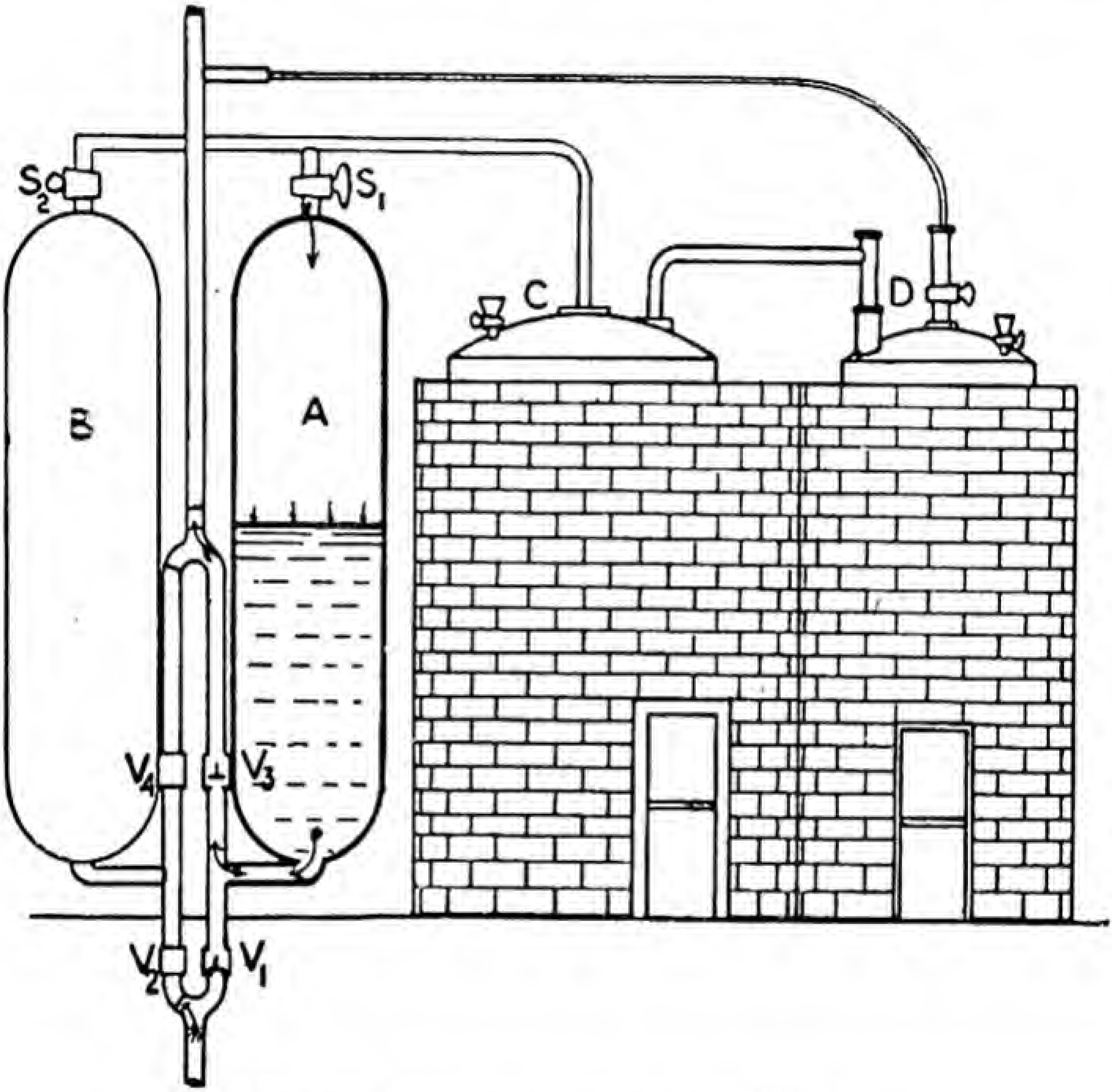


FIG. 333.—Savery's pumping engine (1698).

only could be attained. The principle of Savery's engine is used in the modern pulsometer with the details considerably modified.

The pulsometer.—In the modern pulsometer, constructed by the Pulsometer Engineering Co., Ltd., there are two vessels *A* (Fig. 334), furnished with suction valves *E* near the bottom, and discharge valves *F* shown dotted. Steam is admitted from a supply pipe at the top and will enter either vessel depending on the position of the ball valve *I*. On entering the vessel, the steam presses on the surface of the water and forces it

through the discharge chamber and valves. This operation is accompanied with very little agitation of the water, and consequently there is little condensation of steam until the water level is so low in the vessel that steam begins to blow through the horizontal orifice leading to the discharge valves. Condensation then occurs actively, and a partial vacuum is formed in the vessel which has just been emptied. The sudden reduction of pressure causes the ball valve *I* to roll over and thus shut off this vessel from steam and to open the other, where a process similar to that above explained begins. Meanwhile the partial vacuum in the first vessel enables a further supply of water to enter it through the suction valves. A constant flow of water is obtained practically and the pump is automatic. Small adjustable air valves are fitted near the top of each chamber; these enable the speed of the pump to be adjusted. Owing to the ease with which the pulsometer can be set up (it will work suspended from a crane chain) and to its power of dealing with dirty water, it has found much favour with contractors.

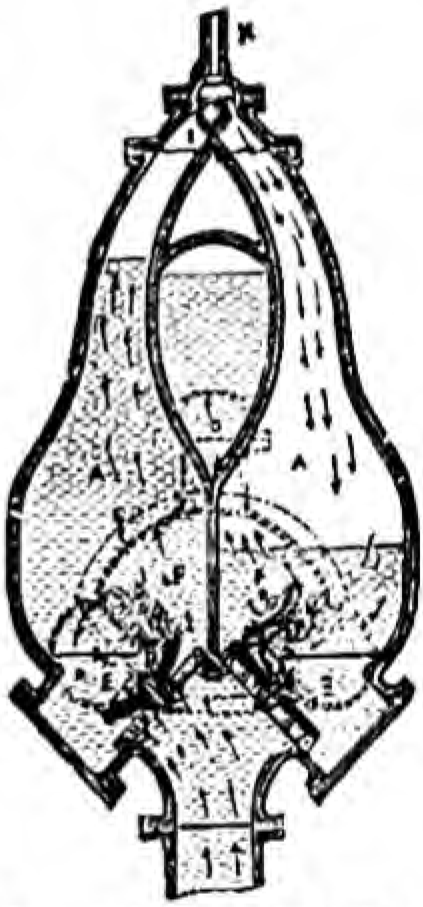


FIG. 334.—Sectional of a pulsometer (1907).

Newcomen's atmospheric engine.—Newcomen in 1705 devised the first practical steam engine containing a cylinder and piston. Fig. 335 shows a drawing of this engine, from which its action will be understood.[1] *A* is the cylinder, arranged vertically, having its upper end open, and having four orifices at the bottom. *B* is the piston attached by means of a chain to a beam *R* vibrating about *T*. To the other end of the beam is attached the pump rod *U*, leading downwards to the pump at the bottom of the mine to be drained. The cylinder *A* is connected to a boiler *K* by a steam pipe *L* furnished with a valve *M*; the steam supplied was at a pressure only slightly greater than that of the atmosphere. *N* is an eduction pipe fitted with a non-return valve at *O*; the object of

[1] Fig. 335 is reproduced by permission from a paper on the "Newcomen Engine," by Mr. Henry Davey, Proc. Inst. Mech. Eng., Oct. 1903.

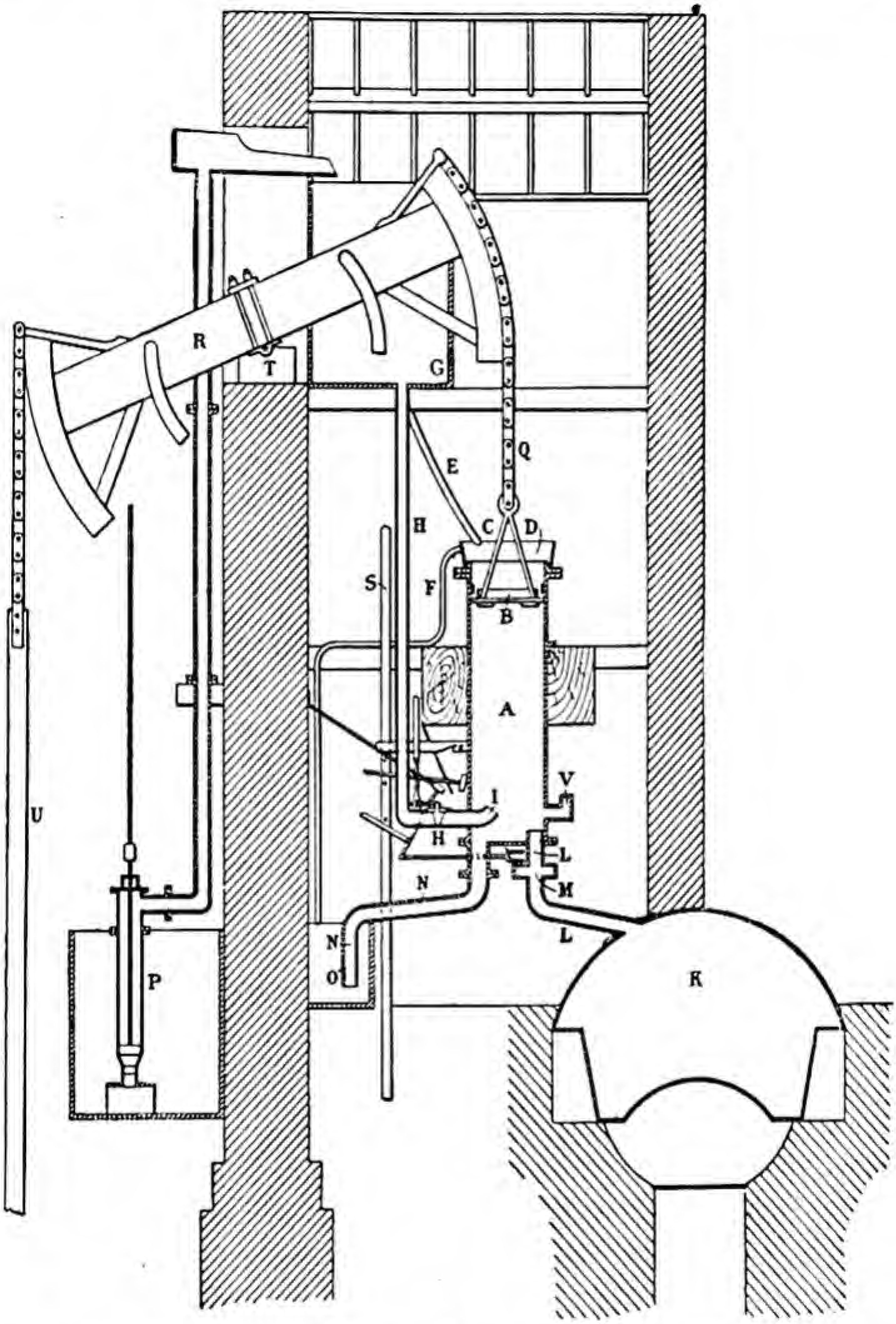


FIG. 335.—Newcomen's atmospheric pumping engine, from a drawing dated 12th January, 1826.

this pipe is to drain off water from the cylinder. *H* is an injection pipe, fitted with a valve near the place where it is connected to the cylinder, and leads from a cistern *G*, from which water may be

injected to the cylinder. *V* is a snifting valve opening outwards; steam may be blown through the cylinder and thereby drive out any contained air through *V*. *S* is a vertical rod, called a plugtree, operated from the beam, and its vertical motion is used to open and close the steam and injection valves. *P* is a pump driven by the beam for keeping the cistern *G* supplied with water. As the cylinder *A* was not bored out, an impossible operation at that time, water was led through a pipe *E* to the piston, and served to prevent leakage of air past the piston into the cylinder. The up stroke of the piston was accomplished by the action of the superior weight of the pump rod *U*.

Action of Newcomen's engine.—The piston being at the top of the stroke, steam is blown through to get rid of air in the cylinder. Then the steam valve *M* is closed, and the injection valve *H* is opened, thus causing a jet of water to play into the cylinder, with the consequent condensation of the steam and the production of a partial vacuum. The superior pressure of the atmosphere on the piston top now urges it downwards, the pump rod *U* rising at the same time, and so operating the pump bucket below. The steam valve *M* is now opened again, admitting steam to the cylinder and thereby equilibrating the pressures on the top and bottom of the piston, thus enabling the superior weight of the pump rod to take effect and perform the up stroke of the piston. During this stroke the water of condensation and the injection water are discharged through the pipe *N*. When at work the steam and injection valves are operated automatically by the engine. It will be noticed that the pressure of the steam was not utilised for doing useful work, but that of the atmosphere, hence the name—**Atmospheric Engine.**

Many engines of the Newcomen type, having cylinders up to 75″ diameter and 8 or 9 feet long, were constructed and put to work for pumping water and winding coal from mines. There is a Newcomen engine, erected in 1810, still employed for winding coal at the Farme Colliery, Rutherglen, near Glasgow. The average pressure on the pistons of these engines was about 7 lbs. per square inch.

James Watt.—In 1763-64, James Watt, who at that time held the post of mathematical instrument maker at the University of Glasgow, had occasion to repair a model of Newcomen's engine belonging to the Natural Philosophy Department. When the

model was set to work he was struck with the enormous loss of steam due to the alternate heating and cooling of the cylinder. Watt observed that the cylinder of the model, being small, exposed more cooling surface to the steam proportionally than would be the case in large engines; that the brass model cylinder conducted heat better than the larger ones of iron; and that to prevent the vacuum being destroyed by re-evaporation, the contents of the cylinder must be cooled to such an extent that much greater condensation occurred on the fresh steam being admitted.

Watt's separate condenser.—In 1765 it occurred to Watt that a hot cylinder and a good vacuum could be obtained if condensation of the steam were effected in a separate condenser. On communication being opened between the cylinder containing steam and another vessel which was exhausted of air and other fluids, the steam, as an expansible fluid, would immediately rush into the empty vessel, and continue to do so until it had established an equilibrium; and if that vessel were kept very cool, by an injection or otherwise, more steam would continue to enter until the whole were condensed. A model was constructed to test this, and was found to give satisfactory results. It is interesting to note that this model was fitted with the first surface condenser, although on the large scale jet condensers were used on account of the bad water with which engines were frequently supplied. An air pump was fitted in order to draw off water and air from the condenser.

In the attempt to keep the cylinder as hot as the entering steam, Watt clothed his cylinders with non-conducting materials, and also fitted steam jackets round the cylinders, which jackets were kept supplied with steam from the boiler.

It is interesting to notice that the same problems of cylinder condensation and re-evaporation, and of the action of the cylinder walls which led to Watt's first improvements on the steam engine are still the subject of investigation by engineers.

Watt's single-acting engine.—In Watt's earliest engines (1769) the action was obtained, as in the Newcomen engine, by the production of a partial vacuum beneath the piston. Condensation of the steam was effected in a separate condenser. The top of the cylinder was closed, and steam at boiler pressure was admitted to the space above the piston, thus taking the place of the atmosphere in the Newcomen engine. The object of this was to get rid of the

cooling action of the air which occurred in the upper portion of the cylinder in the Newcomen engine. The piston rod passed through a hole in the top cover of the cylinder, and was rendered steam-tight by means of a stuffing-box. Wax, tallow, or other grease was used to keep the piston steam-tight and to lubricate it, and the cylinder was jacketed with steam from the boiler, the

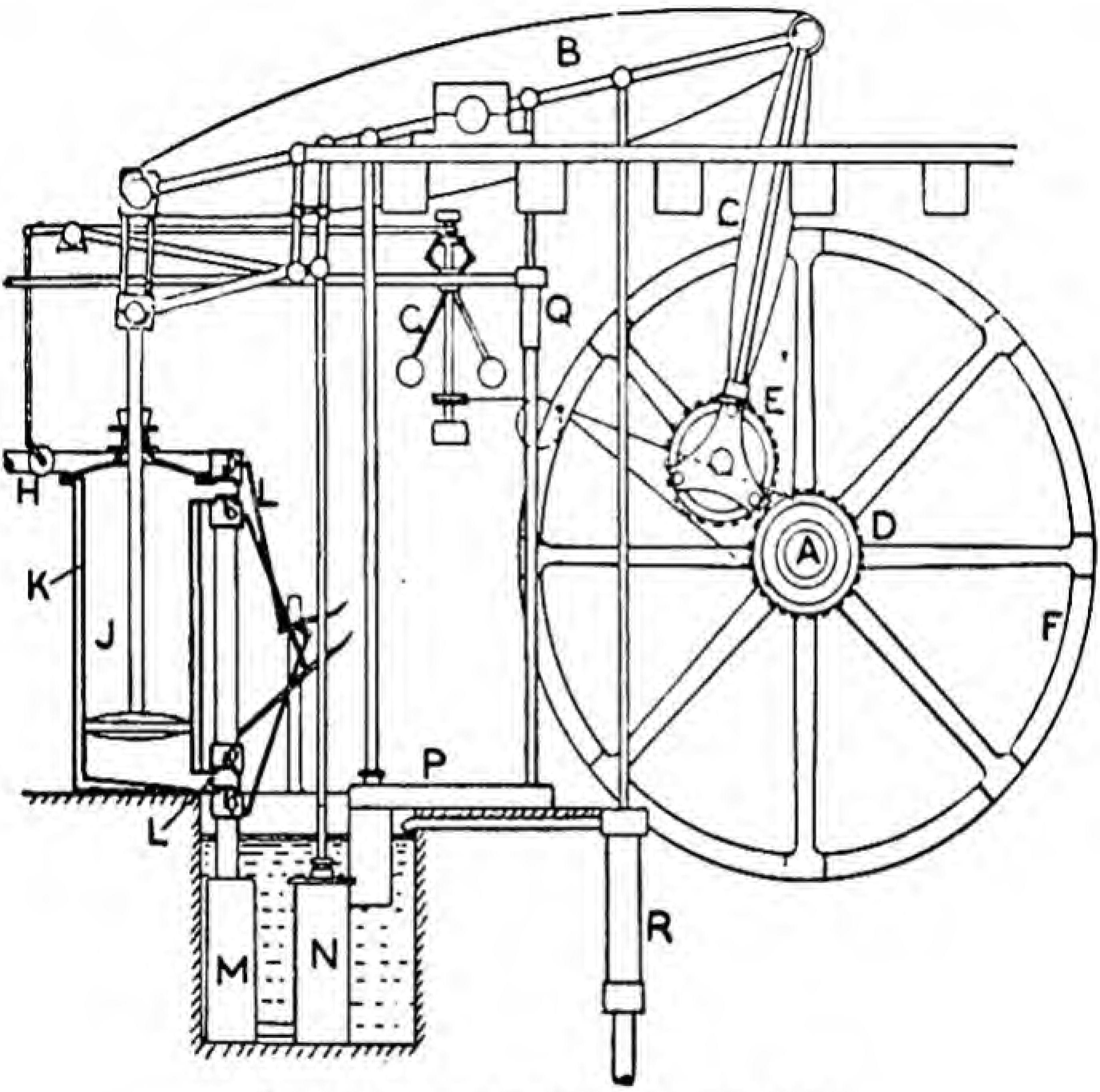


FIG. 336.—Watt's double-acting engine (1782).

outside of the outer cylinder being clothed with wood or other non-conductor of heat.

Watt's double-acting engine.—In this engine, patented in 1782, Watt introduced for the first time the principle of allowing the steam to act alternately on the bottom and top of the piston, the side opposite that on which the steam is acting at any time being in communication with the condenser. This dispensed, for the first time, with the necessity of having a balance weight for

drawing the piston to the top of the stroke, and permitted the piston rod to both pull and push. Watt also introduced in this engine his idea of admitting the steam during a portion only of the stroke, then cutting off further supply and completing the stroke by utilising the expansive action of the steam. Further evidence of Watt's ingenuity will be found by reference to Fig. 336, in which his double-acting engine is illustrated.

Sun-and-planet motion.—In this engine, as in some of Watt's earlier single-acting engines, a rotating shaft *A* (Fig. 336) is introduced, driven from the beam *B* by a connecting rod *C*. Watt devised the crank, but as a prior patent had been taken out, he used his sun-and-planet motion instead until the crank patent had expired. The sun-and-planet motion shown in the figure consists of a toothed wheel *D* secured to the shaft, and another toothed wheel *E* of equal size secured to the end of the connecting rod and gearing with the first. The connecting rod end was guided to move in a circle round the centre of the shaft. With this arrangement, two revolutions of the shaft were secured for each double stroke (one up, one down) of the piston. The crank is universally used now.

Watt's parallel motion.—To enable the piston rod to push as well as pull, Watt applied the parallel motion which now bears his name. This arrangement also served to guide the upper end of the piston rod in a straight line. Its simplest form consists of two equal links *AB*, *CD* (Fig. 337), swinging about centres *A* and *D*, their ends *B* and *C* being connected by a link. If we first put *AB* and *CD* horizontal so that *BC* is vertical, and then move the links up or down, we see that *B* deviates to one side of the vertical and *C* deviates an equal amount to the other side of the vertical, so that *P*, the centre of *BC*, will remain in the vertical. Therefore, for some distance on either side of the mean position of the mechanism,

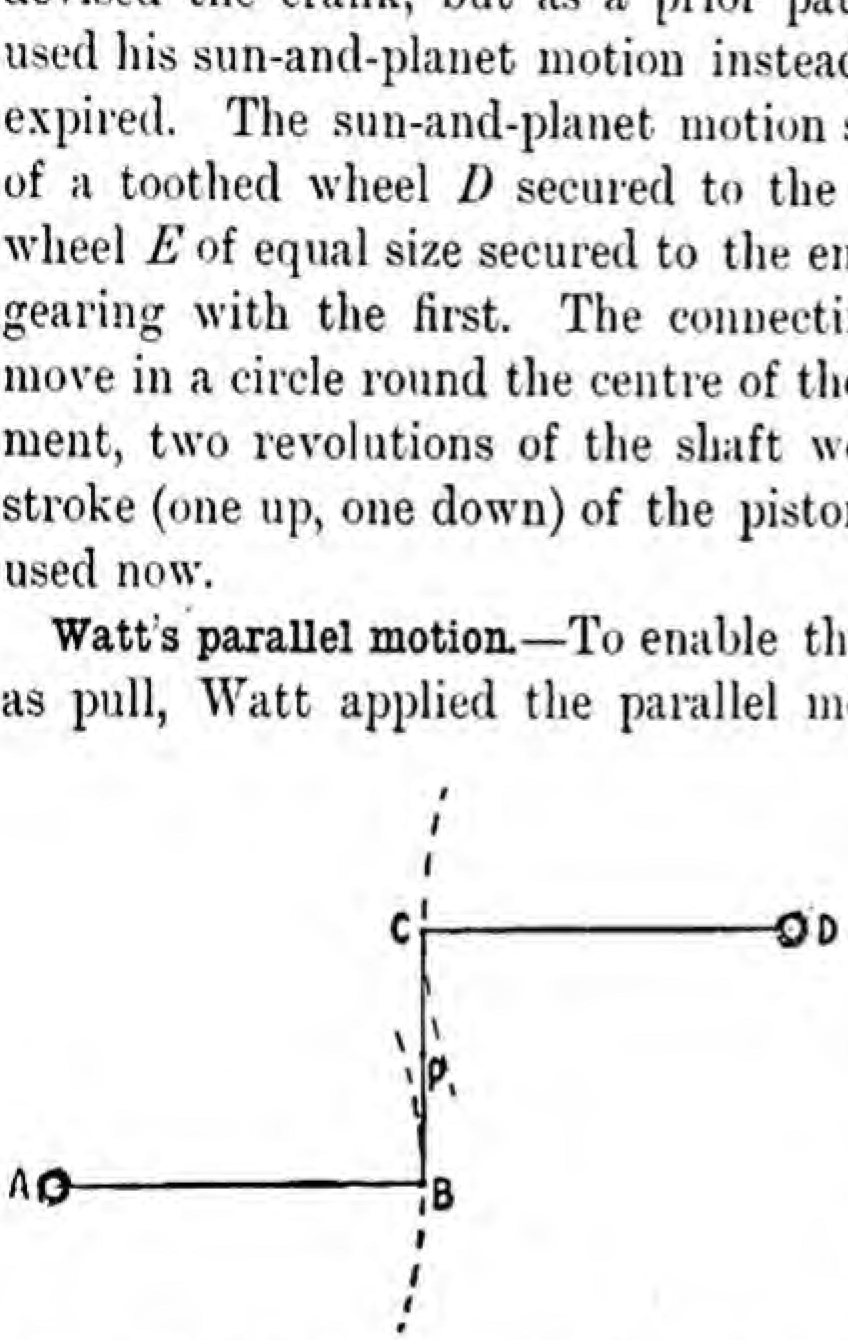


FIG. 337.—Watt's parallel motion having equal arms.

P will remain in a straight line, and this point may be attached to the top of the piston rod so as to guide it in a straight line.

If *AB* and *CD* are unequal (Fig. 338), the deviations of *B* and *C* will be nearly inversely proportional to *AB* and *CD*; of course, it is evident that the deviation of *B* is made greater by decreasing the radius *AB*. If, therefore, *P* is to move in a straight line, *BP* and *PC* must be inversely proportional to *AB* and *CD*, that is,

$$BP : PC = CD : AB.$$

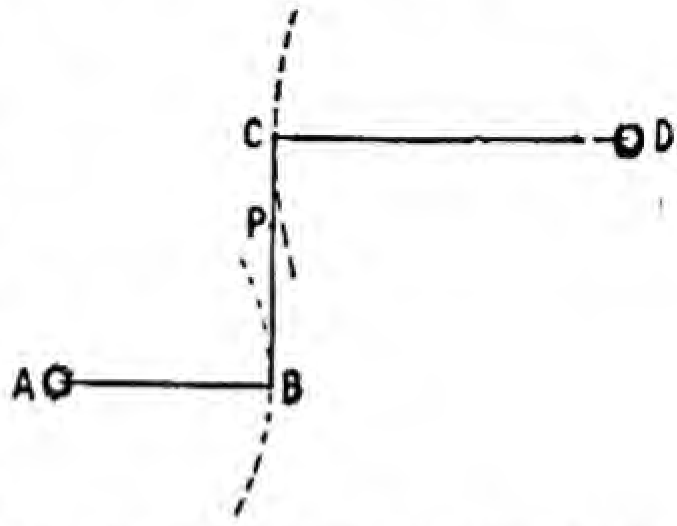


FIG. 338.—Watt's parallel motion having unequal arms.

The arrangement may be extended to give a second point moving in a straight line. In Fig. 339, *DE* represents one half of the beam of the engine, *D* being its fixed centre of vibration. *A* is another fixed centre, and the links *AB*, *BC*, *CD* form the simple parallel motion described above. Supposing *C* to be the middle point of *DE*, and *AB* to be equal to *DC*, then *P*, the centre of the link *BC*, will move in a straight line. Adding the links *FE*=*BC* and *FB*=*EC*, so that *ECBF* is a parallelogram in all positions of the mechanism, it will be found that *F* also moves in a straight line parallel to the line of motion of *P*.

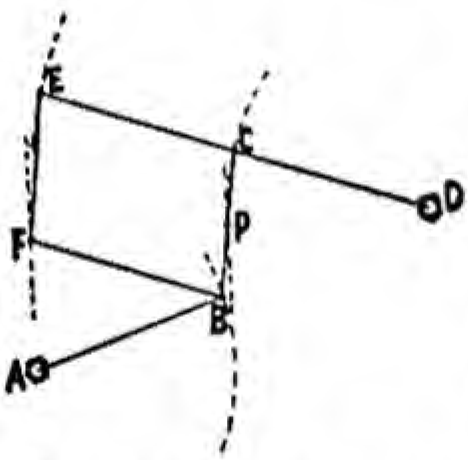


FIG. 339.—Watt's parallel motion for giving straight line motion to two points.

Watt used this latter arrangement, the piston rod being attached to *F*, and the rod for driving the air pump being attached to *P*. The motion is still used for beam engines and also for guiding the pencil in Richard's indicator.

To secure steady motion, a flywheel *F* (Fig. 336) was mounted on the shaft, which, by its inertia, prevented jerky action; and, to keep the speed of the engine fairly constant, Watt applied a centrifugal governor *G* (previously used for controlling the supply of water to water wheels). The governor consists of two heavy balls attached to arms jointed to a vertical shaft driven by the

engine. Should the speed of rotation increase, the balls would move outwards and *vice versâ*. This motion was transmitted by rods and levers to a throttle valve *H* placed in the main steam supply pipe, and so arranged that outward motion of the governor balls tended to close the valve, inward motion opening it. Thus, on an increase of speed taking place, the governor automatically reduced the steam supply and so diminished the speed again to the normal. Too low a speed in the engine caused the governor to supply more steam to the cylinder by opening the throttle valve wider.

The centrifugal governor is still the most trustworthy means we have of controlling the speed of an engine and is used extensively in many different forms.

Action in Watt's double-acting engine.—The cylinder, *J*, is shown in section in Fig. 336, and is fitted with a steam jacket *K*, and a stuffing-box in the top cover for rendering the piston rod steam-tight. Steam and exhaust valves are fitted for each end of the cylinder at *L*, *L*. These are worked by levers and rods operated by the rod which works the air pump. The condenser *M* and the air pump *N* are placed in a well, kept supplied with water by a pump *R*. The injection water required for the condenser is drawn from the well. The water is delivered from the air pump to a hot well *P*, from which the feed pump *Q* draws the boiler feed water. All the pumps are driven by rods from the beam. As has been mentioned already, this engine is double acting; the steam and exhaust valves are opened at the proper instants so as to produce expansive working by tappets on the air pump rod.

High pressure engines.—Owing to the danger of explosion incurred by faulty boiler construction, Watt never used pressures much higher than that of the atmosphere. To Richard Trevethick and others in Cornwall is due the credit of developing engines of higher pressure. Early last century the Cornish engine attained a high efficiency and is still very favourably known. The Cornish engine has a single cylinder, using high-pressure steam expansively, and is fitted with a condenser.

Compound engines.—The first compound engine was patented in 1781 by Hornblower. In this engine two steam cylinders were used, and the steam, after being employed in the first cylinder,

was passed into the other to perform work on the second piston by its expansive action. As a separate condenser and air pump were employed, the engine was an infringement on Watt's patents, and its manufacture was dropped.

The compound engine was revived by Woolf in 1800, but fell into disuse until 1845, when M'Naught modified some existing beam engines by adding another cylinder, and so converting them into compound engines. Such engines at this period began to be used for marine purposes, and in the hands of marine engineers the development was rapid. At the present time marine engines for propeller propulsion are generally of the triple expansion type, the steam being used three times progressively in separate cylinders.

Early marine engines.—The first practical steamer was run in 1802 by William Symington on the Forth and Clyde Canal. This steamer—the *Charlotte Dundas*—was a stern-wheeler, *i.e.* had one paddle wheel working at the stern. Owing to injury to the banks of the canal, the idea was not further developed.

In 1807, Robert Fulton in America fitted the *Clermont* with engines constructed by Boulton & Watt, and ran her successfully. The *Comet*, constructed in 1812 by Henry Bell on the Clyde, was fitted with side-paddles, and was also successful. The invention of the screw-propeller was later, and this means of propulsion was perfected by Ericsson.

Early locomotive engines.—Richard Trevethick in 1802 patented the first steam carriage to run successfully on rails. This engine attained a speed of 5 miles per hour on the Merthyr Tydvil Railroad in South Wales in 1805. In 1829, the *Rocket*, constructed by the Stephensons, by its success in the trials against other competitors on the Stockton and Darlington Railroad, ensured the adoption of locomotive haulage in preference to horse or cables driven by stationary engines. The most important points about the construction of the *Rocket* were the introduction of the steam blast in the chimney for urging the fires, and, for the first time, the use of a multitubular boiler.

EXERCISES ON CHAPTER XXII.

1. Give sketches and explain the action of Savery's engine.

2. Describe the working of the pulsometer. Give sketches.

3. Sketch the cylinder of Newcomen's engine and as much of the mechanism as will enable you to give a description of the working of the engine.

4. Mention Watt's principal improvements in the steam engine.

5. Sketch the cylinder of Watt's double-acting engine. Explain how the steam was distributed.

6. What are the functions of Watt's parallel motion? Explain how two points may be obtained in the mechanism to each of which a piston rod may be connected.

7. Why did Watt's invention of the condenser effect a great economy? Why does condensation take place in the cylinders of modern engines, and how do we attempt to get rid of it? **1897.**

MISCELLANEOUS EXERCISES

FROM RECENT BOARD OF EDUCATION EXAMINATION PAPERS.

1. Describe clearly, with sketches, the working of any single cylinder direct acting non-condensing engine with slide valve and eccentric. Do not give too much detail, but show that you understand how the piston and stuffing-box are made steam-tight; how the piston is fastened to the rod; how the ends of the connecting-rod are made; the action of the governor and of the flywheel. 1897.

2. Sketch and describe the construction of the air pump bucket with its valves and packing, and show how it is worked in connection with a jet condenser. Of what materials are the body of the bucket and of the valves respectively made? 1897.

3. Explain and show, with sketches, the construction and action of the force-pump employed for feeding the water into a boiler when an injector is not used. Sketch also in section the "clack" or non-return valve attached to the boiler. How is the pump prevented from forcing water into the boiler when the engine is running, but a supply of water is not required? 1897.

4. Describe and sketch a Lancashire boiler and its seating. What precautions are taken in stoking to prevent smoke from the furnace of such a boiler? How are the ends strengthened and how are they fastened to the shell? 1897.

5. Describe with sketches the bed or frame of any engine with which you are acquainted.

If you choose either—

(1) A large or small stationary engine, horizontal or vertical;

(2) A locomotive engine;

(3) A marine engine;

(4) A gas or oil engine;

sketch carefully how the cylinder is attached to the frame, and how the slide or slipper is guided in the engine you select; show also the

crank shaft bearing, and how the frame is itself attached to, or supported from, the ground or to the frame of the ship. If you are better acquainted with the construction of a steam turbine or an impulse wheel, describe and sketch one of these instead. 1898.

6. Describe with sketches a boiler of any kind. You need not show any fittings. What are the most important things to remember in connection with the furnace and flue parts? 1898.

7. Answer only *one* of the following questions, *A*, *B*, or *C*:

A. Sketch and describe the staying of the top and sides of a locomotive fire-box, and how the fire-bars are supported.

B. Sketch and describe the construction of the front end plate of either a two-flued Lancashire boiler or a marine boiler (not a water tube boiler), and show how it is connected with the shell plates, and how it is otherwise strengthened or stayed.

C. Show by sketches how the piston-rod and connecting-rod are attached to the crosshead. With a crank of one foot and connecting-rod 5 feet, find by construction the distance of the piston from the near end of the stroke when the crank stands at 30° on either side of each dead point position. 1899.

8. Answer only *one* of the following questions, *A*, *B*, or *C*:

A. Sketch in position in the frame and describe any construction of axle-box of a locomotive engine with which you are acquainted, and show the arrangement of the springs.

B. Sketch and describe a tube igniter, and how the timing valve is worked, in, say, a 20 horse-power gas engine.

C. Sketch the main casting of a large jacketed cylinder, and describe clearly how the cylinder liner and valve seat are attached. 1899.

9. What additional parts are required in order to convert a non-condensing into a condensing engine? Under what circumstances is it better to use a condensing engine? When is it necessary to use a surface condenser? How is a surface condenser constructed? 1899.

10. Describe, with sketches, *either* a gas or oil engine, and show by a diagram how it uses the Otto cycle of operations. Sketch the cylinder showing piston, water jacket, valves, shape of clearance space, and how the exhaust is provided for. 1899.

11. Try only *one* of the following (*a*), (*b*), or (*c*):

(*a*) Describe with sketches the crank pin end of any connecting-rod.

(*b*) Describe with sketches any form of locomotive regulator valve to admit steam from the boiler to the cylinder steam chests.

(*c*) Sketch in section a gas engine cylinder, showing the valves and piston. 1900.

12. Try only *one* of the following (*a*), (*b*), or (*c*):

(*a*) Describe with sketches a Parsons steam turbine. State why it is necessary to make the steam go in series through many elementary turbines.

(*b*) Describe a mechanical stoker and how it acts. Under what circumstances is its use perferable to hand firing.

(*c*) Sketch in section half the crank-shaft of an inside cylinder locomotive, describing the construction of the crank and driving wheel, and showing also the two eccentric discs. 1900.

13. Describe, with sketches, only *one* of the following (*a*), (*b*), (*c*), or (*d*):

(*a*) Any kind of crosshead, showing ends of piston rod and connecting rod and guide.

(*b*) A gas or oil engine cylinder, showing valves and piston.

(*c*) A surface condenser, showing the stays and the attachment of the tubes.

(*d*) An air pump, showing foot, bucket, and delivery valves. 1901.

14. Describe, with sketches, only *one* of the following (*a*), (*b*), (*c*), or (*d*):

(*a*) Locomotive cylinders, and how they are fastened to the side frames.

(*b*) A steam or gas engine governor, and how it regulates.

(*c*) A spirit or oil engine for a motor car, showing how it drives the the car and how it works.

(*d*) The frame of a marine engine, showing how the pumps are worked. 1901.

15. Answer only *one* of the following (*a*), (*b*), or (*c*). Describe how you would experimentally determine:

(*a*) How the pressure and temperature of steam depend upon one another. Why must there be no air present?

(*b*) The calorific power of any kind of burning gas.

(*c*) The latent heat of steam. 1901.

16. Describe, with sketches, only *one* of the following (*a*), (*b*), (*c*), or (*d*):

(*a*) A piston slide valve and its seat, showing packing and ports.

(*b*) Any engine, steam, spirit, or oil used on motor cars.

(*c*) Any link motion or other reversing gear to work a slide valve with which you are acquainted. State exactly what is the effect of altering the gear.

(*d*) Either a Geipel or Sirius or Turnbull or Lancaster steam trap. 1902.

17. Describe, with sketches, only *one* of the following (*a*), (*b*), or (*c*):

(*a*) A double-ended cylindric marine boiler; the usual positions of joints of plates and of stays to be indicated. Where and why is leakage probable under forced draught?

(*b*) Any water-tube boiler; the general construction to be clearly shown: some one part shown in good detail and more carefully described.

(*c*) A steam boiler for a traction engine or a motor car, the fuel being oil or spirit. Describe carefully any appliance necessary in this boiler which is not usually found on a stationary boiler. 1902.

18. Describe, with good sketches, some one important detail of a modern steam or internal combustion engine with which you are well acquainted. If, for example, the crank pin and the end of the connecting rod be shown, it is of no use merely indicating the existence of a bolt and nut; the bolt and nut, and the method of locking the nut, must be clearly shown. Again, it is no use making a sketch of so much of any engine that details cannot be clearly sketched. For example, a whole governor with its gear would be too much, but certain parts may be chosen. 1903.

19. Describe, with good sketches, some one important part of any kind of boiler. For example: a fitting like a safety valve; the staying of the fire-box crown of a locomotive; the arrangement of a furnace; a feed water heater; gauge glass and connections. (The remarks in the previous question apply here also.) 1903.

20. In connection with the steam or gas or oil or spirit engine work with which you are acquainted there is testing of some sort to be done requiring careful measurement of work or heat. For example: finding the calorific power of coal, gas, or oil; finding the latent heat of steam; or how its pressure depends upon the temperature; or finding the wetness of steam during an engine test; comparing the power of an engine and the quantity of heat or of steam, gas, or oil raised per hour. Describe, with sketches, some one such test. 1903.

21. Answer and illustrate by good sketches either (*a*) or (*b*), but not both:

(*a*) In reference to any modern example of a steam, gas, oil, or spirit engine you like to select, how is leakage prevented past the piston? also past the piston rod if a steam engine is chosen?

(*b*) Describe the construction of a steam engine cylinder, showing the ports and the steam inlet and outlet, but omitting the covers of the cylinder and steam chest. 1904.

22. Answer and illustrate by good sketches either (*a*) or (*b*), but not both, in reference to the type of modern steam boiler with which you are best acquainted:

(*a*) Show the arrangement for feeding the boiler with water under pressure.

(*b*) Explain how the steam is brought to the cylinder as dry as possible. Describe a valve for shutting off the supply of steam. 1904.

23. Describe how you would test a boiler for strength, or an indicator for accuracy, or a feed-water meter. Choose only one of these. 1904.

24. What occurs to the coal and its constitutents in the furnace of any boiler? Choose some boiler with which you are well acquainted. What becomes of the heat developed? Trace the products of combustion along the flues as they get cooler, and say what is the nature and state of these products and why the heat is leaving them. 1904.

25. Describe, with sketches, one, and only one, of the following (*a*), (*b*), (*c*), (*d*), or (*e*):

(*a*) The piston of a large steam engine, its packing and fastening to the piston rod.

(*b*) The piston, piston rod, and crosshead of a locomotive.

(*c*) The cylinder of a gas engine.

(*d*) The vanes and mouthpieces of an impulse steam turbine.

(*e*) The cylinder, valves, and igniting arrangement of a petrol engine. 1905.

26. Describe, with sketches, one, and only one, of the following (*a*), (*b*), (*c*), or (*d*):

(*a*) The smoke box of a locomotive showing exhaust steam pipe and cylinder fastenings.

(*b*) A marine safety valve, or a dead-weight safety valve, showing seating block.

(*c*) The general arrangement of any water-tube boiler, some one detail being entered into fully.

(*d*) A Bourdon pressure gauge. 1905.

27. Describe, with good sketches, one, and only one, of the following (*a*), (*b*), (*c*), or (*d*):

(*a*) The crank shaft bearing of a horizontal or vertical engine.

(*b*) The crank axle of an inside cylinder locomotive.

(*c*) The piston of a gas or petrol engine, showing the packing, and the pin to which the connecting rod is attached.

(*d*) The rotating part of a Parsons or other steam turbine, showing how the vanes are fixed.

28. Describe, with good sketches, one, and only one, of the following (*a*), (*b*), (*c*), (*d*), or (*e*):

(*a*) A steam stop valve of the screw-down type.

(*b*) A locomotive regulator valve of any type.

(*c*) Two forms of boiler stays, stating the use of each.

(*d*) The front plate of a Lancashire, Cornish, or return tube marine boiler, showing how the boiler shell is attached.

(*e*) The carburettor of a petrol or oil engine. 1906.

29. Sketch and describe briefly the construction of a piston, showing how it is made steam-tight. Sketch a gland and stuffing-box, and the crank pin end of a connecting rod. 1898.

30. Why do we regulate an engine with both a flywheel and a governor? Explain clearly how each effects the regulation. 1900.

31. What is meant by "Sensible Heat," "Latent Heat," and "Total Heat of Evaporation"? Calculate the total heat in British Thermal Units required to convert 30 lbs. of water at 62° F. into steam at a temperature of 212° F.

If a pound of coal develops 14,000 units of heat during its combustion, how many pounds of coal would be required to convert the 30 lbs. of water into steam under the above conditions, if there was no loss of heat in the operation? 1896.

32. Describe carefully how you would measure the pressure of steam at various temperatures. Show, roughly, what kind of curve you would obtain if you expressed your results on squared paper. 1898.

33. The heat required to convert a pound of water at 0° C. into a pound of steam at θ° C. is

$$H = 606 \cdot 5 + 0 \cdot 305\theta.$$

How much is this if the steam is at 180° C.? How much of this is latent heat? If the water had been at 40° C. to begin with, what total heat was needed? 1905.

34. State the following amounts of energy in foot-pounds:

(*a*) A weight of one ton which may fall vertically 10 feet.

(*b*) 3 lbs. of water raised 20 degrees Centigrade.

(*c*) One horse-power hour.

(*d*) 3 watts for 200 hours. 1901.

35. What do we mean by working steam expansively? Steam at 60 lbs. per sq. in. absolute is used in a cylinder whose stroke is 2 feet, and expansion begins when one-quarter of the stroke has been performed. What are probable pressures at half, three-quarters, and the end of the stroke? Show your answers in a diagram. Find the average pressure. 1898.

36. Steam enters a cylinder at 180 lbs. pressure per square inch (absolute); is cut off at one-third of the stroke, and expands according to the law "pv constant." Find the average pressure (absolute) during the forward stroke, neglecting clearance. If the back pressure is 17 lbs. (absolute) per square inch, what is the average effective pressure? If the area of the cross-section of the cylinder is 112 square inches and the stroke is 24 inches, what work is done in one stroke? 1900.

37. A steam engine has a steam cylinder of 20 inches in diameter, the crank measures 18 inches from the centre of crank-shaft to centre of crank-pin, the engine runs at 85 revolutions per minute, and the mean effective pressure of steam on the piston is 28 lbs. per square inch. Find the indicated horse-power of the engine. 1896.

38. What information ought to be found written beside an indicator diagram? How would you proceed to find the indicated horse-power? The average breadth of the two diagrams on one card is 1·56 inches. Scale, $\frac{1}{32}$. Piston, 12 inches diameter. Crank, 1 foot. 110 revolutions per minute. Find the indicated horse-power. What is the actual horse-power given out by the crank-shaft likely to be? 1897.

39. Steam enters a cylinder at 150 lbs. (absolute) per square inch. It is cut off at one-fourth of the stroke and expands according to the law "pv constant." Find the average pressure (absolute) in the forward stroke. If the back pressure is 17 lbs. (absolute) per square inch, what is the average effective pressure? If the area of the cross-section of the cylinder is 126 square inches, and the crank is 11 inches long, what work is done in one stroke? Neglecting clearance and condensation, what volume of steam enters the cylinder per stroke? 1901.

40. Using the formula $pV^{1\cdot0646}=479$, find the volume of 1 lb. of the steam admitted to the cylinder in Question 39. What weight of steam is actually admitted to that cylinder per stroke? Or you may use instead of the formula, the following information and squared paper: Steam at 150 lbs. pressure is at 181° C., and the following numbers are known:

Temperature - - - - - - -	175° C.	180° C.	185° C.
Volume in cubic feet of 1 lb. of steam -	3·419	3·065	2·756

1901.

41. Make a sketch and describe the construction of an eccentric sheave and strap. Show the position of the crank-shaft through the eccentric, and indicate on your sketch the throw of the eccentric. Name the materials of which the several parts of the eccentric are made. 1896.

42. Describe and sketch a slide valve, describe how it distributes the steam, and how it is worked. What are meant by the terms lap, half-travel, inside lap, and advance? 1898.

43. Sketch a simple slide valve showing cylinder ports, and the valve-chest. What do we mean by *outside lap* of a valve; *inside lap*; *advance*; *lead*? What is the effect of each of these on the indicator diagram? 1900.

44. If cut-off takes place on both sides of a piston when the crank makes an angle of 90° with the dead point; (1) assuming connecting rod infinitely long; (2) assuming connecting rod four times length of crank; find in each case for each side of piston the fraction of stroke at which cut-off takes place. 1901.

45. If a piston with its rod weighs 290 lbs., and if at a certain instant when the resultant total force due to the steam pressure is 7 tons, the piston has an acceleration of 420 ft. per second per second in the same direction, what is the actual force acting on the crosshead? 1900.

46. One pound of fuel contains 0·8 lb. of carbon and 0·15 lb. of hydrogen and no free oxygen or nitrogen: what weight of oxygen is needed for complete combustion? What weight of air? 1905.

47. Why is priming such a great evil? What is the cause of the condensation of steam in a cylinder before cut-off? How do we try to diminish it? 1898.

48. Explain the difference between a simple non-condensing engine, a condensing engine, and a compound non-condensing engine. Give outline sketches of the general arrangement in a horizontal engine of each of the three classes. 1896.

49. An engine whose speed and cut-off do not alter, uses W lbs. steam per hour when its actual horse-power is P, and W and P have been carefully measured during three long tests.

P	152	110	56
W	3190	2630	1850

What is the probable W when P is 125 horse-power? In each of the four cases find the steam used per horse-power hour. 1903.

COURSE OF LABORATORY WORK.

General Instructions.—Two Laboratory Note-books are required; in one rough notes of the experiments should be made, and in the other a fair copy of them in ink should be entered.

Before commencing any experiment, make sure that you understand what its object is; also the construction of the apparatus and instruments employed.

Reasonable care should be exercised in order to avoid damage to apparatus, and to secure fairly accurate results.

In writing up the results, enter the notes in the following order:

(1) The title of the experiment and the date on which it was performed.

(2) Sketches and descriptions of any special apparatus or instruments used.

(3) The object of the experiment.

(4) Dimensions, weights, etc., required for working out the results; from these values calculate any constants required.

(5) Log of the experiment, entered in tabular form where possible, together with any remarks necessary.

(6) Work out the results of the experiment and tabulate them where possible.

(7) Plot any curves required.

(8) Work out any general equations required.

Notes should not be left in the rough form for several days; it is much better to work out the results and enter them directly after the experiments have been performed.

TEMPERATURE. EXPANSION.

1. **Determination of the freezing point error of a thermometer.**—Perform the experiment as directed (p. 21).

2. **Determination of the boiling point error of a thermometer.**—Follow the directions given (p. 22).

3. **Determination of the graduation errors of a thermometer.**—Perform the experiment as indicated (p. 23).

4. **Expansion of a metal tube.**—Perform the experiment as directed (p. 27).

5. **Expansion of water.**—Perform the experiment as directed (p. 27).

HEAT AND ITS MEASUREMENT.

6. **Temperature of mixtures of water.**—Perform the experiment, following the directions given (p. 31).

7. **Convection currents.**—Perform the experiment as directed (p. 34). Give a sketch showing by arrows the directions of the currents.

8. **Temperature of mixtures.**—Perform the experiment as indicated (p. 35).

9. **Specific heat of solids.**—Determine the specific heats of the samples of iron, copper, brass, etc., supplied to you (p. 36).

10. **Relative values of non-conducting materials.**—Perform the experiment as directed (p. 37).

PROPERTIES OF GASES.

11. **Common barometer.**—Perform the experiment as directed (p. 43).

12. **Pressure of the atmosphere.**—Perform the experiment and make the required calculations (p. 44).

13. **Verification of Boyle's law.**—Perform the experiment as directed (p. 47).

PROPERTIES OF STEAM.

14. **Latent heat of ice.**—Perform the experiment as directed (p. 59).

15. **Latent heat of steam at atmospheric pressure.**—Perform the preliminary experiment (p. 60).

16. **Latent heat of steam at atmospheric pressure.**—Perform the experiment as given (p. 63), taking the precautions stated and making the required corrections.

17. **Superheated steam.**—Perform the experiment as directed (p. 63).

18. **Relation of p and t for saturated steam.**—Perform the experiment as indicated (p. 65).

USE OF INDICATORS AND BRAKES.

19. **Attachment of the indicator and gear.**—Carry out the instructions given (pp. 97, 104).

20. **Attachment of the brake.**—Carry out as directed (pp. 103, 104).

21. **Taking indicator diagrams.**—Follow the directions given (pp 98, 104).

22. **Brake horse power.**—Carry out as directed (pp. 103, 104).

SLIDE VALVES.

23. **Valve for non-expansive working.**—Perform the experiment as explained (p. 111).

24. **Valve for expansive working.**—Perform the experiment as directed (p. 115).

25. **Valve having inside lap.**—Perform the experiment as indicated (p. 116).

26. **Practical case of a slide valve.**—Perform the experiment, following the instructions given (p. 117).

27. **Link motion.**—Perform the experiment as directed (p. 122).

MISCELLANEOUS.

28. **Inertia of a vibrating weight.**—Perform the experiment as directed (p. 133).

29. **Gyrostatic effect of a revolving wheel.**—Perform the experiment as explained (p. 142).

30. **Effect of fluid pressure on a curved flattened tube.**—Perform the experiment as directed (p. 183).

31. **Heating value of coal.**—Test the sample of coal supplied to you, following the instructions (p. 231).

STEAM ENGINE TRIALS.

32. **Mechanical efficiency of the H.P. side of a compound engine.**—Follow the directions laid down (p. 358).

33. **Mechanical efficiency of the L.P. side of a compound engine.**—Follow the directions given (p. 359).

34. **Mechanical efficiency of a coupled compound engine.**—Carry out as directed (p. 359).

35. **Steam consumption of an engine.**—Follow the directions laid down (p. 360).

STEAM BOILER TRIALS.

36. **Evaporative capacity of a steam boiler.**—Carry out the test as directed (p. 364). In working out the results, follow the plan shown in the example (p. 366).

37. **Dryness fraction of steam.**—Use the separating calorimeter and follow the directions given (p. 370).

38. **Dryness fraction of steam.**—Use the throttling calorimeter and follow the directions given (p. 371).

INTERNAL COMBUSTION ENGINE TRIALS.

39. **Heat disposal in a gas engine.**—Follow the methods described in Chapter XXI.

40. **Heat disposal in an oil engine.**—Follow the methods explained in Chapter XXI.

MATHEMATICAL TABLES.

USEFUL CONSTANTS.

1 inch	$=2{\cdot}54$ centimetres $=25{\cdot}4$ millimetres.
1 metre	$=39{\cdot}37$ inches.
5280 feet	$=1$ mile.
1 kilometre	$=0{\cdot}621$ mile.
1 square inch	$=6{\cdot}45$ square centimetres.
1 square metre	$=1550$ square inches.
1 cubic inch	$=16{\cdot}39$ cubic centimetres.
1 cubic metre	$=61{,}025$ cubic inches $=1{\cdot}308$ cubic yards.
1 litre	$=1000$ cubic centimetres $=1{\cdot}7617$ pint.
1 gallon	$=0{\cdot}1605$ cubic foot $=4{\cdot}541$ litres.
1 radian	$=57{\cdot}3$ degrees.
π	$=3{\cdot}1416$.

1 knot = 6080 feet per hour.
60 miles per hour = 1 mile per minute = 88 feet per second.
The value of g at London = 32·182 feet per sec. per sec.

One pound avoirdupois = 7000 grains = 453·6 grams.
One kilogram = 2·205 pounds.
One gallon of pure water at 62° F. weighs 10 lbs.
One cubic foot of pure water at 62° F. weighs 62·3 lbs.
Weight of 1 lb. in London = 445,000 dynes.
One cubic foot of air at 0° C. and 1 atmosphere pressure weighs 0·0807 lb.
One cubic foot of hydrogen at 0° C. and 1 atmosphere pressure weighs 0·00559 lb.

1 atmosphere $= 14{\cdot}7$ lbs. per square inch
$= 2116$ lbs. per square foot
$= 10^6$ dynes per square centimetre nearly.

1 kilogram per square centimetre $= 14{\cdot}22$ lbs. per square inch.

A column of mercury 760 millimetres ($= 30$ inches) high produces a pressure at its base of 1 atmosphere.

A column of water $2{\cdot}3$ feet high produces a pressure at its base of 1 lb. per sq. inch.

1 foot-pound	$= 1{\cdot}3562 \times 10^7$ ergs.
1 metre-kilogram	$= 7{\cdot}235$ foot-pounds.
1 horse-power	$= 33{,}000$ foot-pounds per minute $= 746$ watts.
1 horse-power-hour	$= 33{,}000 \times 60$ foot-pounds.
Volts $\times$ ampères	$=$ watts.
1 electrical unit	$= 1000$ watt-hours.

1 B.T.U.	$= \frac{5}{9}$ lb.-degree-Cent. unit
	$= 252$ gram-calories.

Absolute temperature $\tau = t^\circ \text{C.} + 273{\cdot}7,$
$= t^\circ \text{F.} + 461.$

Regnault's $H = 606{\cdot}5 + {\cdot}305\, t^\circ$ C. (Centigrade units.)
$= 1082 + {\cdot}305\, t^\circ$ F. (Fahrenheit units.)

Joule's equivalent to suit Regnault's $H = \begin{cases} 774 \text{ ft.-lbs.} = 1 \text{ Fah. unit.} \\ 1393 \text{ ft.-lbs.} = 1 \text{ Cent. unit.} \end{cases}$

$pV^{1{\cdot}0646} = 479.$ (Zeuner's formula.)

$pV^{\frac{17}{16}} = 479.$ (Rankine's formula.)

$\log_{10} p = 6{\cdot}1007 - \frac{B}{\tau} - \frac{C}{\tau^2}.$ (Rankine's formula.)

where $p =$ pressure in lbs. per sq. inch.
$V =$ volume in cubic feet per pound of steam.
$\tau =$ absolute temperature Centigrade.

$\log_{10} B = 3{\cdot}1812.$
$\log_{10} C = 5{\cdot}0881.$

To convert common into Napierian logarithms, multiply by $2{\cdot}3026$.
The base of the Napierian logarithms is $e = 2{\cdot}7183$.

Napierian or Hyperbolic Logarithms.

No.	Log.	No.	Log.
1·11	0·1043	4	1·3864
1·143	0·1335	5	1·6095
1·25	0·2231	6	1·7919
1·33	0·2853	7	1·9459
1·43	0·3576	8	2·0795
1·6	0·4700	9	2·1971
1·66	0·5068	10	2·3026
2	0·6931	11	2·3979
2·5	0·9162	12	2·4850
2·66	0·9784	13	2·5649
3	1·0986	14	2·6390
3·33	1·2029	15	2·7081

This table has been calculated for those numbers only which most commonly occur as ratios of expansion.

Pressure, lbs. per sq. inch.	Temperature, F.	Volume of 1 lb. in cubic feet.	Heat Units (Fahrenheit).			Pressure, lbs. per sq. inch.	Temperature, F.	Volume of 1 lb. in cubic feet.	Heat Units (Fahrenheit).		
			H	h	$(H-h) = L$				H	h	$(H-h) = L$
1	102	334.2	1113.0	70.1	1043	115	337.8	3.821	1185.0	309.5	875.5
2	126.3	173.2	1120.4	94.4	1026	120	341.0	3.671	1185.9	312.8	873.2
3	141.6	118.0	1124.9	109.9	1015	125	344.1	3.534	1186.9	316.0	870.9
4	153.1	89.80	1128.6	121.4	1007	130	347.1	3.406	1187.8	319.0	868.7
5	162.3	72.50	1131.4	130.7	1001	135	349.9	3.287	1188.7	322.1	866.6
6	170.1	61.10	1133.8	138.6	995.2	140	352.8	3.177	1189.5	325.0	864.6
7	176.9	53.00	1135.9	145.4	990.5	145	355.5	3.074	1190.4	327.8	862.6
8	182.9	46.60	1137.7	151.5	986.2	150	358.2	2.978	1191.2	330.6	860.6
9	188.3	41.82	1139.4	156.9	982.4	155	360.7	2.888	1192.0	333.2	858.7
10	193.2	37.80	1140.9	161.9	979.0	160	363.3	2.803	1192.7	335.9	856.9
15	213.0	25.87	1146.9	181.9	965.0	165	365.7	2.724	1193.5	338.4	855.1
20	227.9	19.72	1151.4	197.0	954.4	170	368.2	2.649	1194.2	340.9	853.3
25	240.0	15.99	1155.1	209.3	945.8	175	370.5	2.578	1194.9	343.4	851.6
30	250.2	13.48	1158.3	219.7	938.5	180	372.8	2.510	1195.6	345.8	849.9
35	259.2	11.66	1161.0	228.8	932.1	185	375.1	2.447	1196.3	348.1	848.2
40	267.1	10.29	1163.4	236.9	926.5	190	377.3	2.386	1197.0	350.4	846.6
45	274.3	9.21	1165.6	244.3	921.3	195	379.4	2.328	1197.7	352.7	845.0
50	280.8	8.34	1167.6	251.0	916.6	200	381.6	2.273	1198.3	354.9	843.4
55	286.9	7.63	1169.4	257.1	912.3	205	383.7	2.221	1199.0	357.1	841.9
60	292.5	7.03	1171.2	262.9	908.2	210	385.7	2.171	1199.6	359.2	840.4
65	297.8	6.52	1172.8	268.3	904.5	215	387.7	2.124	1200.2	361.3	838.9
70	302.7	6.09	1174.3	273.4	900.9	220	390.0	2.07	1200.8	—	—
75	307.4	5.70	1175.7	278.2	897.5	230	394	1.99	1201.9	—	—
80	311.8	5.37	1177.0	282.7	894.3	240	397.5	1.91	1203.0	—	—
85	316.0	5.07	1178.3	287.0	891.3	250	401	1.84	1204.0	—	—
90	320.0	4.81	1179.5	291.2	888.4	260	404.5	1.77	1205.0	—	—
95	323.9	4.57	1180.7	295.1	885.6	270	408	1.71	1206.0	—	—
100	327.6	4.356	1181.8	298.9	882.9	280	411	1.65	1207.0	—	—
105	331.1	4.161	1182.9	302.6	880.3	290	414.4	1.60	1207.9	—	—
110	334.5	3.984	1184.0	306.1	877.9	300	417.5	1.55	1208.9	—	—

This table is extracted, with permission, from *The Steam Engine and Gas and Oil Engines*, Prof. John Perry, F.R.S. (Macmillan.)

Properties of Steam. (Centigrade Units.)

Temperature, C.	Pressure. Lbs. per sq. inch.	Pressure. Lbs. per sq. foot.	Volume of 1 lb. in cubic feet. V_s	Heat Units (Cent.). H	Heat Units (Cent.). h sensible Heat	Heat Units (Cent.). $(H-h) = L$	Heat Units (Cent.). External work during evaporation.	Heat Units (Cent.). Intrinsic energy of 1 lb. of steam.
0	0·085	12·27	3398	606·5	0·00	606·5	29·92	576·6
5	0·122	17·62	2412	608·0	5·00	603·0	30·51	577·5
10	0·173	24·92	1736	609·5	10·00	599·5	31·06	578·4
15	0·241	34·77	1268	611·1	15·00	596·0	31·65	579·4
20	0·333	47·87	936·9	612·6	20·01	592·6	32·19	580·4
25	0·452	65·06	700·8	614·1	25·02	589·1	32·73	581·4
30	0·607	87·40	530·7	615·6	30·03	585·6	33·30	582·3
35	0·806	116·1	405·9	617·2	35·04	582·1	33·84	583·4
40	1·06	152·6	313·6	618·7	40·05	578·6	34·34	584·4
45	1·38	198·6	244·6	620·2	45·07	575·1	34·87	585·3
50	1·78	256·0	192·5	621·7	50·09	571·7	35·37	586·3
55	2·27	327·0	152·8	623·3	55·11	568·2	35·85	587·4
60	2·88	414·3	122·3	624·8	60·13	564·7	36·37	588·4
65	3·62	520·6	98·7	626·3	65·17	561·1	36·88	589·4
70	4·51	649·1	80·23	627·8	70·20	557·6	37·39	590·4
75	5·58	803·3	65·64	629·4	75·24	554·1	37·85	591·5
80	6·86	987·6	54·06	630·9	80·28	550·6	38·32	592·6
85	8·38	1206	44·81	632·4	85·33	547·1	38·79	593·6
90	10·16	1463	37·36	633·9	90·38	543·6	39·23	594·7
95	12·26	1765	31·34	635·5	95·44	540·0	39·70	595·8
100	14·70	2116·4	26·43	637·0	100·5	536·5	40·15	596·8
105	17·53	2524	22·40	638·5	105·6	533·0	40·58	597·9
110	20·80	2994	19·08	640·9	110·6	529·4	41·02	599·0
115	24·54	3534	16·32	641·6	115·7	525·8	41·40	600·2
120	28·83	4152	14·04	643·1	120·8	522·3	41·83	601·3
125	33·71	4854	12·12	644·6	125·9	518·7	42·23	602·4
130	39·25	5652	10·51	646·1	131·0	515·1	42·64	603·5
135	45·49	6551	9·147	647·7	136·1	511·6	43·00	604·7
140	52·52	7563	7·995	649·2	141·2	508·0	43·40	605·8
145	60·40	8698	7·009	650·7	146·3	504·4	43·76	606·9
150	69·21	9966	6·168	652·2	151·5	500·8	44·14	608·1
155	79·03	11380	5·446	653·8	156·5	497·2	44·48	609·3
160	89·86	12940	4·827	655·3	161·7	493·5	44·83	610·5
165	101·9	14680	4·290	656·8	166·9	489·9	45·20	611·6
170	115·1	16580	3·823	658·3	172·0	486·3	45·50	612·8
175	129·8	18690	3·419	659·9	177·2	482·7	45·86	614·0
180	145·8	20990	3·065	661·4	182·4	479·0	46·17	615·2
185	163·3	23520	2·756	662·9	187·6	475·3	46·52	616·4
198	182·4	26270	2·482	664·4	192·8	471·7	46·80	617·6
195	203·3	29270	2·242	666·0	198·0	468·0	47·10	618·9
200	225·9	32520	2·031	667·5	203·2	464·3	47·42	620·1
205	250·3	36050	1·843	669·0	208·4	460·6	47·68	621·3
210	276·9	39870	1·676	670·6	213·7	456·9	47·97	622·6
215	305·5	43990	1·529	672·1	218·9	453·2	48·27	623·8

This table is extracted, with permission, from *The Steam Engine and Gas and Oil Engines*, Prof. John Perry, F.R.S. (Macmillan.)

Boiling points of water at pressures near standard atmospheric pressure.

(The pressures are given in mm. of mercury at a temperature of 0° C. situated at the sea level in latitude 45°.)

Pressure mm.	Temp. deg. Cent.	Pressure mm.	Temp. deg. Cent.	Pressure mm.	Temp. deg. Cent.	Pressure mm.	Temp. deg. Cent.
733	98·9939	745	99·4449	757	99·8897	769	100·3286
734	99·0318	746	·4822	758	·9265	770	·3649
735	·0695	747	·5194	759	·9633	771	·4012
736	·1073	748	·5567	760	100·0000	772	·4374
737	·1449	749	·5938	761	·0367	773	·4736
738	·1826	750	·6310	762	·0733	774	·5098
739	·2202	751	·6681	763	·1099	775	·5459
740	·2577	752	·7051	764	·1465	776	·5820
741	·2953	753	·7421	765	·1830	777	·6180
742	·3327	754	·7791	766	·2194	778	·6540
743	·3702	755	·8160	767	·2559	779	·6900
744	·4075	756	·8529	768	·2923	780	·7259

This table is taken, with permission, from Edser's *Heat for Advanced Students.* (Macmillan.)

Table of coefficients of linear expansion.

(These are given as the increase in length which a bar of unit length undergoes when heated through one degree Fahrenheit.)

Steel alloyed with 36 % nickel - -	0·000000483
Wrought-iron and mild steel - -	0·00000673
Cast-iron - - - - - -	0·0000063
Copper - - - - - - -	0·0000096
Zinc - - - - - - -	0·0000162
Brass - - - - - - -	0·0000105
Phosphor-bronze - - - - -	0·0000107

Table of Specific Heats.

Aluminium	0·2022
Copper	0·09232
Iron	0·1098
Lead	0·0315
Nickel	0·1084
Platinum	0·03147
Tin	0·0559
Zinc	0·0935

LOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170						4	9	13	17	21	26	30	34	38
						0212	0253	0294	0334	0374	4	8	12	16	20	24	28	32	36
11	0414	0453	0492	0531	0569						4	8	12	15	19	23	27	31	35
						0607	0645	0682	0719	0755	4	7	11	15	19	22	26	30	33
12	0792	0828	0864	0899	0934						3	7	11	14	18	21	25	28	32
						0969	1004	1038	1072	1106	3	7	10	14	17	20	24	27	31
13	1139	1173	1206	1239	1271						3	7	10	13	16	20	23	26	30
						1303	1335	1367	1399	1430	3	7	10	13	16	19	22	25	29
14	1461	1492	1523	1553	1584						3	6	9	12	15	19	22	25	28
						1614	1644	1673	1703	1732	3	6	9	12	15	17	20	23	26
15	1761	1790	1818	1847	1875						3	6	9	11	14	17	20	23	26
						1903	1931	1959	1987	2014	3	6	8	11	14	17	19	22	25
16	2041	2068	2095	2122	2148						3	5	8	11	14	16	19	22	24
						2175	2201	2227	2253	2279	3	5	8	10	13	16	18	21	23
17	2304	2330	2355	2380	2405						3	5	8	10	13	15	18	20	23
						2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648						2	5	7	9	12	14	16	19	21
						2672	2695	2718	2742	2765	2	5	7	9	11	14	16	18	21
19	2788	2810	2833	2856	2878						2	4	7	9	11	13	16	18	20
						2900	2923	2945	2967	2989	2	4	6	8	11	13	15	17	19
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8

These tables of logarithms and antilogarithms are taken, with permission, from *Machine Construction and Drawing*, by Frank Castle. (Macmillan.)

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·00	1000	1002	1005	1007	1009	1012	1014	1016	1019	1021	0	0	1	1	1	1	2	2	2
·01	1023	1026	1028	1030	1033	1035	1038	1040	1042	1045	0	0	1	1	1	1	2	2	2
·02	1047	1050	1052	1054	1057	1059	1062	1064	1067	1069	0	0	1	1	1	1	2	2	2
·03	1072	1074	1076	1079	1081	1084	1086	1089	1091	1094	0	0	1	1	1	1	2	2	2
·04	1096	1099	1102	1104	1107	1109	1112	1114	1117	1119	0	1	1	1	1	2	2	2	2
·05	1122	1125	1127	1130	1132	1135	1138	1140	1143	1146	0	1	1	1	1	2	2	2	2
·06	1148	1151	1153	1156	1159	1161	1164	1167	1169	1172	0	1	1	1	1	2	2	2	2
·07	1175	1178	1180	1183	1186	1189	1191	1194	1197	1199	0	1	1	1	1	2	2	2	2
·08	1202	1205	1208	1211	1213	1216	1219	1222	1225	1227	0	1	1	1	1	2	2	2	3
·09	1230	1233	1236	1239	1242	1245	1247	1250	1253	1256	0	1	1	1	1	2	2	2	3
·10	1259	1262	1265	1268	1271	1274	1276	1279	1282	1285	0	1	1	1	1	2	2	2	3
·11	1288	1291	1294	1297	1300	1303	1306	1309	1312	1315	0	1	1	1	2	2	2	2	3
·12	1318	1321	1324	1327	1330	1334	1337	1340	1343	1346	0	1	1	1	2	2	2	2	3
·13	1349	1352	1355	1358	1361	1365	1368	1371	1374	1377	0	1	1	1	2	2	2	3	3
·14	1380	1384	1387	1390	1393	1396	1400	1403	1406	1409	0	1	1	1	2	2	2	3	3
·15	1413	1416	1419	1422	1426	1429	1432	1435	1439	1442	0	1	1	1	2	2	2	3	3
·16	1445	1449	1452	1455	1459	1462	1466	1469	1472	1476	0	1	1	1	2	2	2	3	3
·17	1479	1483	1486	1489	1493	1496	1500	1503	1507	1510	0	1	1	1	2	2	2	3	3
·18	1514	1517	1521	1524	1528	1531	1535	1538	1542	1545	0	1	1	1	2	2	2	3	3
·19	1549	1552	1556	1560	1563	1567	1570	1574	1578	1581	0	1	1	1	2	2	3	3	3
·20	1585	1589	1592	1596	1600	1603	1607	1611	1614	1618	0	1	1	1	2	2	3	3	3
·21	1622	1626	1629	1633	1637	1641	1644	1648	1652	1656	0	1	1	2	2	2	3	3	3
·22	1660	1663	1667	1671	1675	1679	1683	1687	1690	1694	0	1	1	2	2	2	3	3	3
·23	1698	1702	1706	1710	1714	1718	1722	1726	1730	1734	0	1	1	2	2	2	3	3	4
·24	1738	1742	1746	1750	1754	1758	1762	1766	1770	1774	0	1	1	2	2	2	3	3	4
·25	1778	1782	1786	1791	1795	1799	1803	1807	1811	1816	0	1	1	2	2	2	3	3	4
·26	1820	1824	1828	1832	1837	1841	1845	1849	1854	1858	0	1	1	2	2	3	3	3	4
·27	1862	1866	1871	1875	1879	1884	1888	1892	1897	1901	0	1	1	2	2	3	3	3	4
·28	1905	1910	1914	1919	1923	1928	1932	1936	1941	1945	0	1	1	2	2	3	3	4	4
·29	1950	1954	1959	1963	1968	1972	1977	1982	1986	1991	0	1	1	2	2	3	3	4	4
·30	1995	2000	2004	2009	2014	2018	2023	2028	2032	2037	0	1	1	2	2	3	3	4	4
·31	2042	2046	2051	2056	2061	2065	2070	2075	2080	2084	0	1	1	2	2	3	3	4	4
·32	2089	2094	2099	2104	2109	2113	2118	2123	2128	2133	0	1	1	2	2	3	3	4	4
·33	2138	2143	2148	2153	2158	2163	2168	2173	2178	2183	0	1	1	2	2	3	3	4	4
·34	2188	2193	2198	2203	2208	2213	2218	2223	2228	2234	1	1	2	2	3	3	4	4	5
·35	2239	2244	2249	2254	2259	2265	2270	2275	2280	2286	1	1	2	2	3	3	4	4	5
·36	2291	2296	2301	2307	2312	2317	2323	2328	2333	2339	1	1	2	2	3	3	4	4	5
·37	2344	2350	2355	2360	2366	2371	2377	2382	2388	2393	1	1	2	2	3	3	4	4	5
·38	2399	2404	2410	2415	2421	2427	2432	2438	2443	2449	1	1	2	2	3	3	4	4	5
·39	2455	2460	2466	2472	2477	2483	2489	2495	2500	2506	1	1	2	2	3	3	4	5	5
·40	2512	2518	2523	2529	2535	2541	2547	2553	2559	2564	1	1	2	2	3	4	4	5	5
·41	2570	2576	2582	2588	2594	2600	2606	2612	2618	2624	1	1	2	2	3	4	4	5	5
·42	2630	2636	2642	2649	2655	2661	2667	2673	2679	2685	1	1	2	2	3	4	4	5	6
·43	2692	2698	2704	2710	2716	2723	2729	2735	2742	2748	1	1	2	3	3	4	4	5	6
·44	2754	2761	2767	2773	2780	2786	2793	2799	2805	2812	1	1	2	3	3	4	4	5	6
·45	2818	2825	2831	2838	2844	2851	2858	2864	2871	2877	1	1	2	3	3	4	5	5	6
·46	2884	2891	2897	2904	2911	2917	2924	2931	2938	2944	1	1	2	3	3	4	5	5	6
·47	2951	2958	2965	2972	2979	2985	2992	2999	3006	3013	1	1	2	3	3	4	5	5	6
·48	3020	3027	3034	3041	3048	3055	3062	3069	3076	3083	1	1	2	3	4	4	5	6	6
·49	3090	3097	3105	3112	3119	3126	3133	3141	3148	3155	1	1	2	3	4	4	5	6	6

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·50	3162	3170	3177	3184	3192	3199	3206	3214	3221	3228	1	1	2	3	4	4	5	6	7
·51	3236	3243	3251	3258	3266	3273	3281	3289	3296	3304	1	2	2	3	4	5	5	6	7
·52	3311	3319	3327	3334	3342	3350	3357	3365	3373	3381	1	2	2	3	4	5	5	6	7
·53	3388	3396	3404	3412	3420	3428	3436	3443	3451	3459	1	2	2	3	4	5	6	6	7
·54	3467	3475	3483	3491	3499	3508	3516	3524	3532	3540	1	2	2	3	4	5	6	6	7
·55	3548	3556	3565	3573	3581	3589	3597	3606	3614	3622	1	2	2	3	4	5	6	7	7
·56	3631	3639	3648	3656	3664	3673	3681	3690	3698	3707	1	2	3	3	4	5	6	7	8
·57	3715	3724	3733	3741	3750	3758	3767	3776	3784	3793	1	2	3	3	4	5	6	7	8
·58	3802	3811	3819	3828	3837	3846	3855	3864	3873	3882	1	2	3	4	4	5	6	7	8
·59	3890	3899	3908	3917	3926	3936	3945	3954	3963	3972	1	2	3	4	5	5	6	7	8
·60	3981	3990	3999	4009	4018	4027	4036	4046	4055	4064	1	2	3	4	5	6	6	7	8
·61	4074	4083	4093	4102	4111	4121	4130	4140	4150	4159	1	2	3	4	5	6	7	8	9
·62	4169	4178	4188	4198	4207	4217	4227	4236	4246	4256	1	2	3	4	5	6	7	8	9
·63	4266	4276	4285	4295	4305	4315	4325	4335	4345	4355	1	2	3	4	5	6	7	8	9
·64	4365	4375	4385	4395	4406	4416	4426	4436	4446	4457	1	2	3	4	5	6	7	8	9
·65	4467	4477	4487	4498	4508	4519	4529	4539	4550	4560	1	2	3	4	5	6	7	8	9
·66	4571	4581	4592	4603	4613	4624	4634	4645	4656	4667	1	2	3	4	5	6	7	9	10
·67	4677	4688	4699	4710	4721	4732	4742	4753	4764	4775	1	2	3	4	5	7	8	9	10
·68	4786	4797	4808	4819	4831	4842	4853	4864	4875	4887	1	2	3	4	6	7	8	9	10
·69	4898	4909	4920	4932	4943	4955	4966	4977	4989	5000	1	2	3	5	6	7	8	9	10
·70	5012	5023	5035	5047	5058	5070	5082	5093	5105	5117	1	2	4	5	6	7	8	9	11
·71	5129	5140	5152	5164	5176	5188	5200	5212	5224	5236	1	2	4	5	6	7	8	10	11
·72	5248	5260	5272	5284	5297	5309	5321	5333	5346	5358	1	2	4	5	6	7	9	10	11
·73	5370	5383	5395	5408	5420	5433	5445	5458	5470	5483	1	3	4	5	6	8	9	10	11
·74	5495	5508	5521	5534	5546	5559	5572	5585	5598	5610	1	3	4	5	6	8	9	10	12
·75	5623	5636	5649	5662	5675	5689	5702	5715	5728	5741	1	3	4	5	7	8	9	10	12
·76	5754	5768	5781	5794	5808	5821	5834	5848	5861	5875	1	3	4	5	7	8	9	11	12
·77	5888	5902	5916	5929	5943	5957	5970	5984	5998	6012	1	3	4	5	7	8	10	11	12
·78	6026	6039	6053	6067	6081	6095	6109	6124	6138	6152	1	3	4	6	7	8	10	11	13
·79	6166	6180	6194	6209	6223	6237	6252	6266	6281	6295	1	3	4	6	7	9	10	11	13
·80	6310	6324	6339	6353	6368	6383	6397	6412	6427	6442	1	3	4	6	7	9	10	12	13
·81	6457	6471	6486	6501	6516	6531	6546	6561	6577	6592	2	3	5	6	8	9	11	12	14
·82	6607	6622	6637	6653	6668	6683	6699	6714	6730	6745	2	3	5	6	8	9	11	12	14
·83	6761	6776	6792	6808	6823	6839	6855	6871	6887	6902	2	3	5	6	8	9	11	13	14
·84	6918	6934	6950	6966	6982	6998	7015	7031	7047	7063	2	3	5	6	8	10	11	13	15
·85	7079	7096	7112	7129	7145	7161	7178	7194	7211	7228	2	3	5	7	8	10	12	13	15
·86	7244	7261	7278	7295	7311	7328	7345	7362	7379	7396	2	3	5	7	8	10	12	13	15
·87	7413	7430	7447	7464	7482	7499	7516	7534	7551	7568	2	3	5	7	9	10	12	14	16
·88	7586	7603	7621	7638	7656	7674	7691	7709	7727	7745	2	4	5	7	9	11	12	14	16
·89	7762	7780	7798	7816	7834	7852	7870	7889	7907	7925	2	4	5	7	9	11	13	14	16
·90	7943	7962	7980	7998	8017	8035	8054	8072	8091	8110	2	4	6	7	9	11	13	15	17
·91	8128	8147	8166	8185	8204	8222	8241	8260	8279	8299	2	4	6	8	9	11	13	15	17
·92	8318	8337	8356	8375	8395	8414	8433	8453	8472	8492	2	4	6	8	10	12	14	15	17
·93	8511	8531	8551	8570	8590	8610	8630	8650	8670	8690	2	4	6	8	10	12	14	16	18
·94	8710	8730	8750	8770	8790	8810	8831	8851	8872	8892	2	4	6	8	10	12	14	16	18
·95	8913	8933	8954	8974	8995	9016	9036	9057	9078	9099	2	4	6	8	10	12	15	17	19
·96	9120	9141	9162	9183	9204	9226	9247	9268	9290	9311	2	4	6	8	11	13	15	17	19
·97	9333	9354	9376	9397	9419	9441	9462	9484	9506	9528	2	4	7	9	11	13	15	17	20
·98	9550	9572	9594	9616	9638	9661	9683	9705	9727	9750	2	4	7	9	11	13	16	18	20
·99	9772	9795	9817	9840	9863	9886	9908	9931	9954	9977	2	5	7	9	11	14	16	18	20

ANSWERS.

Chapter II., p. 29.

2. 284° F. **3.** $-40°$ C. **4.** $-459·4°$ F.

5. 34·05″. **6.** 432° F. **8.** 1·505″.

Chapter III., p. 41.

2. 23·55 lb.-degree-Cent. units ; 10,680 gram-calories.

3. 33·12 B.T.U. **5.** 77·1° F. **6.** 64·07° F.

7. 62·9° F. **9.** 140,040 foot-lbs. **10.** 391·5°C.

11. 11,670,000 foot-lbs. ; 15,950,000 foot-lbs. ; 466,900 foot-lbs.

12. 7,795,000 B.T.U.

Chapter IV., p. 56.

3. 0·513 cubic foot. **4.** 405·1 cubic inches.

6. 173·5 cubic feet. **7.** 929·2° C.

Chapter V., p. 72.

4. 66·64 lbs. per sq. inch. **5.** 4·368 cubic feet.

6. $L = 905·5$ B.T.U. ; $H = 1173·5$ B.T.U. **7.** 146,900 B.T.U.

9. 102,660 B.T.U. **10.** 34·6 lbs.

11. 1125·5 B.T.U. ; 12·78 lbs. **12.** 618·3 lb.-degree-Cent. units.

Chapter VI., p. 87.

1. 309·6 metre-kilograms. **2.** 6,930,000,000 foot-lbs.

3. 89 lbs. per sq. inch ; 252,000 foot-lbs.

4. 43·9 lbs. per sq. inch ; 124,100 foot-lbs.

6. 1·667 ; 1·571 ; 1630 cubic inches.

8. (*a*) 1,176,000 foot-lbs. ; (*b*) 3,727,000 foot-lbs. ; (*c*) 11,910,000 foot-lbs. ; (*d*) 1,470,000 foot-lbs. ; (*e*) 1,980,000 foot-lbs. ; (*f*) 2,655,000 foot-lbs.

9. 83·4 lbs. per sq. inch ; 24,020 foot-lbs. ; 0·7 cubic foot ; 34,310 foot-lbs.

10. 2714 cubic inches ; 4·08 per cent. ; 0·192″.

Chapter VII., p. 106.

4. 214·5 I.H.P. **5.** 211·4 I.H.P. **7.** 7·174 B.H.P.

8. 304 B.T.U. per min. **9.** 8·69 I.H.P.; 50,160 foot-lbs.

12. 94,696 foot-lbs.

Chapter VIII., p. 125.

1.

Crank angle	30°	60°	90°	120°	150°
Piston position	0·08	0·29	0·56	0·79	0·95

3. 30°; −6·5°; 126°. **4.** 150°; 330°.

Chapter IX., p. 147.

1. 4600 lb.-feet. **2.** 1000 lbs.; 14·3 sq. inches.

4. 15,081 lbs. **5.** 15,245 lbs. **7.** 3106 lbs.

8. 301·6 feet per sec. per sec; 191·9 feet per sec. per sec.

11. 2917 lb.-feet. **13.** 367·8 foot-tons. **14.** 116 revs. per min.

15. 0·0255 foot-tons; 206·5 foot-tons. **17.** 4·7″. **18.** 4·87″.

21. 4236 lbs. **22.** Centrifugal force $=0·003408\,n^2r$ lbs.; 242; 246.

Chapter XI., p. 202.

2. 6033 lbs.; 1·207 sq. inches; 1·24″. **6.** 38·8″.

13. 141·4 lbs. **15.** 1374 lbs.

Chapter XII., p. 215.

1. 10,450 lbs. per sq. inch. **2.** 0·5″.

7. $d=\frac{15}{16}''$; $p=3\frac{5}{16}''$; 70·6 per cent. **8.** $\frac{23}{32}''$. **9.** $1\frac{3}{8}''$.

Chapter XIII., p. 234.

2. 28 lbs.; 228·3 lbs. **3.** 50 cubic feet.

4. 126 lbs.; 1027 lbs. **5.** 32,480,000 B.T.U.

7. 13,925 B.T.U. **8.** 10·95 lbs.; 22 lbs.

11. 14,505 B.T.U. **12.** 19,040,000 lb.-deg. C.; £7. 10s. 3·6*d*.; £252.

13. 3·47 lbs.; 15·09 lbs.

Chapter XIV., p. 251.

1. 80 per cent. **2.** 73 per cent. **3.** 17,960 B.T.U.

4. 16,450 B.T.U. **5.** 16,490 B.T.U.; 15·4 per cent.

15. 8·76 per cent.

Chapter XV., p. 292.

1. H.P., 60·5 lbs. per sq. inch; L.P., 19·3 lbs. per sq. inch.
6. 6·8.
7. 17,300 lbs.; 11,000 lbs.
18. 8·79 cubic feet of air at a pressure of 2 lbs. per sq. inch.
23. 11·5 lbs. per sq. inch.
26. 273 gallons.

28.

p_1, lbs. per sq. in.	100	80	60
p_m, lbs. per sq. in.	59·7	44·4	29
W, foot-lbs.	143,000	106,500	69,600

29. 3·33 cubic feet;

p.	100	80	60
Weight of steam used per stroke, lbs.	0·765	0·621	0·474
Weight of steam used per rev., lbs.	1·53	1·242	0·948

Chapter XVI., p. 316.

10. 1875 lbs.
11. 121·3 I.H.P.
12. 114,000 lbs.; 12,670 lbs.; 1020 H.P.

Chapter XVIII., p. 373.

3. 27,310 lbs.
4. 22,780 B.T.U.
8. 24·4 lbs
9. 43,125 lbs.;

Output, kilowatts	6000	1200	600
Water per hour per kilowatt	22	44·17	71·9

10. 92·4 per cent.

Chapter XIX., p. 389.

8. 63·4 lbs. per sq. inch.
9. 8800 foot-lbs.; about three quarters, viz. 6600 foot-lbs.

Chapter XXI., p. 425.

3. 93·8 cubic feet.

5. 633 B.T.U.; 492,000 foot lbs.

6. 10·9 per cent.

7. Mech. eff. = 76 per cent.; Heat eff. on I.H.P. = 16·25 per cent.; Heat-eff. on B.H.P. = 12·33 per cent.

9.

Points	*A*	*B*	*C*	*D*
Temp. C.	127°	207°	1328°	687°

Miscellaneous, p. 439.

7. *C.* 0·108 foot from inner, 0·16 foot from outer dead point.

31. 33,500 B.T.U.; 2·39 lbs.

33. 661·4 lb.-deg.-Cent.; 481·4 lb.-deg.-Cent.; 621·4 lb.-deg.-Cent.

34. (*a*) 22,400; (*b*) 83,580; (*c*) 1,980,000; (*d*) 1,590,000; all in foot-lbs.

35. 30; 20; 15; 35·8; all in lbs. per sq. inch.

36. 125·8 lbs. per sq. inch; 108·8 lbs. per sq. inch; 24,370 foot-lbs.

37. 136·2 I.H.P.

38. 75·3 I.H.P.; 60 B.H.P.

39. 89·5 lbs. per sq. inch; 72·5 lbs. per sq. inch; 16,750 foot-lbs.; 0·4 cubic foot.

40. 2·98 cubic feet; 0·134 lb. weight.

44. 0·5; 0·5; 0·56; 0·44.

45. 11,900 lbs.

46. 3·33 lbs.; 14·98 lbs.

49.

P	152	125	110	56
W	3190	2820	2630	1850
Steam per horse-power hour, lbs.	21	22·55	23·9	33·05

INDEX.

PRINTED IN GREAT BRITAIN BY ROBERT MACLEHOSE AND CO. LTD.
THE UNIVERSITY PRESS, GLASGOW.